A STUDY OF AUTOMATIC SHUTTLE LOOM DYNAMICS AND POWER CONSUMPTION OF ITS VARIOUS MECHANISMS

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A STUDY OF AUTOMATIC SHUTTLE LOOM DYNAMICS AND POWER CONSUMPTION OF ITS VARIOUS MECHANISMS

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SUMMARY

In this thesis, an analysis of automatic shuttle loom kinematics, dynamics and power consumption of the three most important mechanisms have been made. These are 1) Beating up, 2) Shedding and 3) Picking.

These mechanisms have been evaluated for a Draper X - 2 loom of 44" reed space. These looms are currently used all over the world and can be considered typical. The kinematic and dynamic equations have been derived by using standard techniques that have been introduced in technical papers by different authors.

After a brief introduction, the equations have been derived and listed. Then computer programs have been evolved and the parameters of interest have been evaluated. It was found that shaking forces and moments transmitted to the frame are large and need to be minimized. Also the power efficiency is rather low.

Then various proposed methods for improving the efficiency of these looms have been introduced and their advantages and disadvantages discussed.

Recommendations for future studies are given.

CHAPTER I

INTRODUCTION

In this thesis, an analysis of the dynamics of some important mechanisms of conventional fly shuttle looms have been made and their energy consumption estimated.

Some new proposed mechanisms are presented and their benefits and disadvantages discussed from the point of view of energy use efficiency, ease of installation and versatility of operation.

Purpose of the Research

The energy use efficiency of automatic fly shuttle looms is low. Basically the principles of automatic shuttle weaving have changed very little since such looms were introduced at the beginning of this century. Mainly this is because automatic shuttle looms are very versatile by nature and can compete with most other types of weaving machines on the basis of production per hour, efficiency of operation and running cost per unit length of fabric produced. Although in recent years, many of these looms have been replaced by other more sophisticated types of weaving machines there still are many automatic shuttle looms in operation.

The following figures(1) indicate the number of looms in place in the U.S.A., at the end of 1972.

Type of Yarn	No. of Looms
Cotton	207100
Woolen and Worsted	5200
Manmade Fibers	111200
Total	323500

Of these, less than 5% were of the shuttleless type. In the world, there were an estimated 2,656,190 cotton looms[2] at the end of 1970. Of these again less than 5% were of the shuttleless type.

In 1933, a study made by Honegger[3] at the Federal Institute of Technology at Zurich, it was found that of the total power consumed by a shuttle loom, only 8% went into the making of the fabric. The remaining 92% was consumed in overcoming frictional forces and in accelerating the shuttle. Thus efficiencywise the shuttle loom is in league with the steam engine.

Over the years, very little research has been carried out to change loom conditions. There are many reasons for this. Primarily loom manufacturers have focused their attention on increasing production and improving the quality of the woven products. Many novel approaches have been made and many types of shuttleless and high speed shuttle looms have evolved. In the wake of the world energy crisis, and the consequent need to reduce energy consumption, the efficiency of weaving machines need to be improved.

To date, not much research has been done on this topic. A few researchers have worked on shuttle propulsion problems, prominent among whom are Catlow[4] and Vincent[5]. Some research has been done on energy consumption by Honegger[3]. But the main purpose of this research was

Table 1. Specifications of Draper X – 2 Loom Used for Experimental Work

Make:	Draper
Model:	X – 2
Reed Width:	44 inches
Motor:	
Make:	Diehl
H.P.	1.0
Volts:	220
Amps:	2.7
Synchronous Speed:	1745 R.P.M.
Clutch:	Friction type

to find a suitable drive for automatic looms and not to improve its efficiency. A mathematical treatise on plain loom action has been introduced by Wilmot[6]. A few other researchers have done some research on speed and torque variation. In the area of loom noise, considerable research has been done and many methods suggested for its control.

CHAPTER II

THEORETICAL ANALYSIS OF PRIMARY FLY SHUTTLE LOOM MECHANISMS

From an energy consumption standpoint, the main loom mechanisms are 1) Beating up, 2) Shedding, and 3) Picking. These three mechanisms consume an estimated 73% of the total power required. For the purpose of this thesis, therefore only these three mechanisms have been studied in depth. A Draper X - 2 model loom has been used for all experimental and theoretical studies; detailed specifications of this loom are given in Table (1).

Beating Up Mechanisms

The beating up mechanism (also called slay mechanism) is a four-bar linkage of Type 1. Figure (1) shows an arbitrary type 1 four-bar linkage with its nomenclature and dimensions. The numerical values of these dimensions for a Draper X - 2 loom are given in Table (2).

In accordance with normal nomenclature, link no. 1 (a_1) is the machine frame. Link No. 2 (a_2) consists of the two cranks connected to the crankshaft, which, in any shuttle loom is normally supported by three to five journal bearings (five in the case of the X - 2 loom). Link No. 3 (a_3) consists of the two connecting rods. Link No. 4 (a_4) is the entire slay and associated machine members pivoted about the bottom rocking shaft.

This four-bar mechanism has been studied in two steps. First,

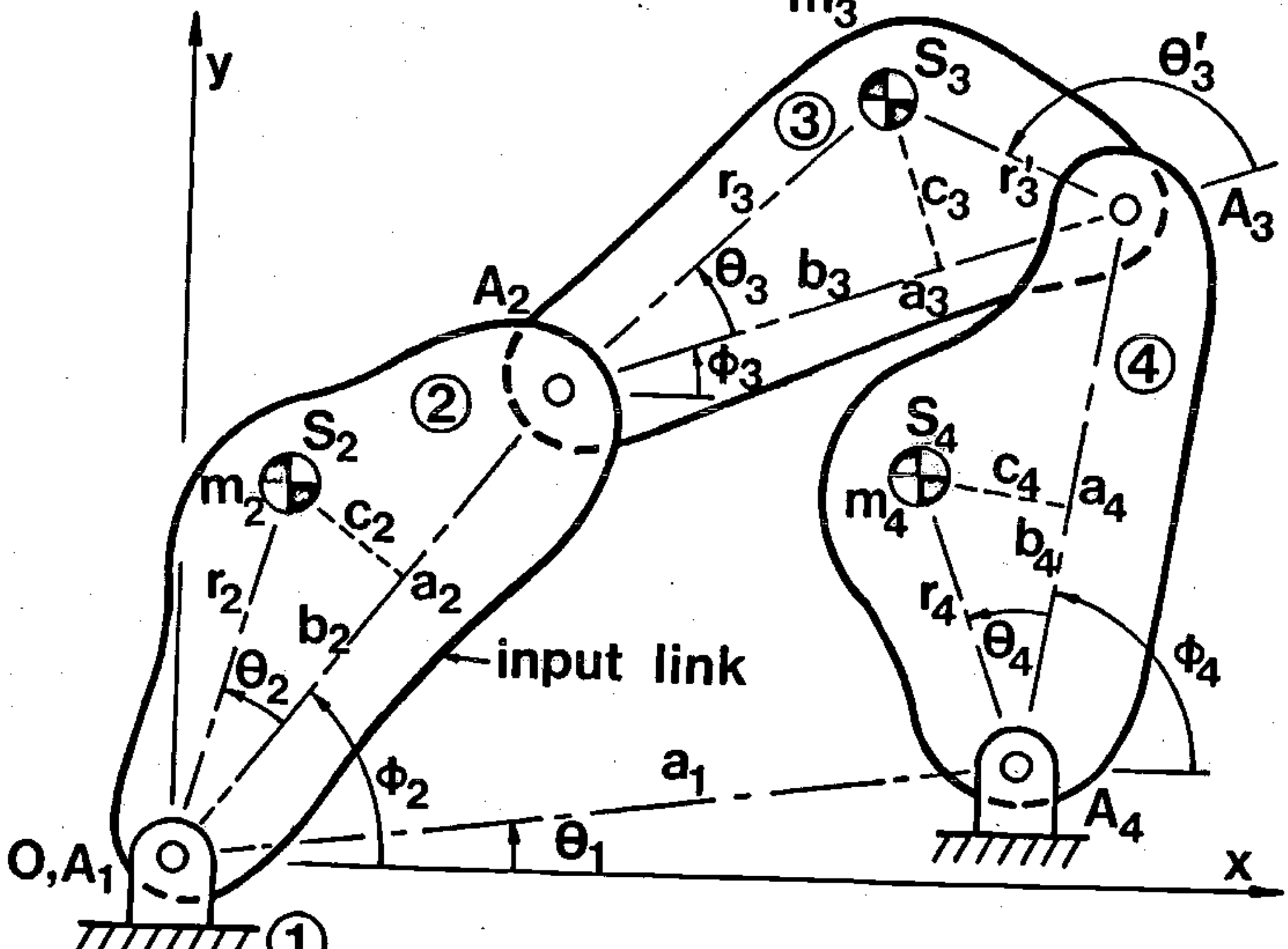


Figure 1. Four-bar Linkage Nomenclature

all the kinematic parameters have been evaluated and then, the forces and moments have been computed. The basic kinematic and dynamic equations developed by Berkof[7,8] for the four-bar linkage have been used. The nomenclature of Figure (1) is explained below.

A_i (i = 1,2,3,4)	=	no. of pin joint or bearing
a_i (i = 1,2,3,4)	=	kinematic length of link i
S_i (i = 2,3,4)	=	centre of mass of link i
r_i (i = 2,3,4)	=	distance from pin joint $A_{(i-1)}$ to centre of mass s_i
θ_1	=	angle between link no. 1 (a_i) and x axis of reference system measured counterclockwise from x axis
θ_i (i = 2,3,4)	=	angle between r_i and a_i measured counterclockwise from a_i
ϕ_i (i = 2,3,4)	=	angular position of link i with respect to x axis of the reference system
m_i (i = 2,3,4)	=	mass of link i
b_i (i = 2,3,4)	=	$r_i \cos \theta_i$ for link i
C_i (i = 2,3,4)	=	$r_i \sin \theta_i$ for link i
r_3'	=	length of A_3S_3
θ_3'	=	angle between A_3S_3 and a_3, measured counterclockwise from a_3
μ	=	transmission angle ($\phi_4 - \phi_3$)

The position of the centre of mass S_i, with respect to pivot $A_{(i-1)}$, for any link i for a certain value of ϕ_i can be expressed as:

$$\xi_i = r_i \cdot \cos(\phi_i + \theta_i)$$

$$\eta_i = r_i \cdot \sin(\phi_i + \theta_i) \qquad (i = 2, 3, 4) \qquad (2.1)$$

The time derivatives of this equation are:

$$\dot{\xi}_i = -\eta_i \dot{\phi}_i$$

$$\dot{\eta}_i = \xi_i \dot{\phi}_i \qquad (2.2)$$

$$\ddot{\xi}_i = -\eta_i \ddot{\phi}_i - \xi \dot{\phi}_i^2$$

$$\ddot{\eta}_i = \xi_i \ddot{\phi}_i - \eta_i \dot{\phi}_i^2 \quad (i = 2, 3, 4) \qquad (2.3)$$

An equation for link angle ϕ_4, as a function of ϕ_2, is derived by writing the vector loop equation for the linkage, separating the cartesian components to give two equations, eliminating ϕ_3 between them by trigonometric substitution, and using trigonometric identities to arrive at a quadratic equation in tan ($\frac{1}{2}\phi_4$).

Thus:

$$\phi_4 = 2 \tan^{-1}\left(\frac{A \pm \sqrt{A^2 + B^2 - C^2}}{B + C}\right) \qquad (2.4)$$

where:

$$A = \sin(\phi_2)$$

Table 2. Detailed Specification of Draper X - 2 Loom

Parameter	Numerical Value
a_1	29.5 inches
a_2	2.7 inches
a_3	10.31 inches
a_4	27.1 inches
r_2	1.424 inches
r_3	5.0 inches
r_4	18.34 inches
0_1	0.0 degrees
0_2	0.0 degrees
0_3	0.0 degrees
0_4	-4.87 degree
$_2$	0.436 slugs
$_3$	0.404 slugs
$_4$	7.932 slugs
b_2	1.424 inches
b_3	5.0 inches
b_4	18.27 inches
c_2	0.0 inches
c_3	0.0 inches
c_4	-1.55 inches
r_3	5.31 inches

Table 2 (cont'd.)

Parameter	Numerical Value
0_3	180 degrees
k_2	2.38 inches
k_3	3.58 inches
k_4	14.0 inches

x - The k_i's have been found out by first finding the moment of inertia of each link by the pendulum method[13].

$$B = \cos(\phi_2) - \frac{a_1}{a_2}$$

$$C = \frac{a_1^2 + a_2^2 - a_3^2 + a_4^2}{2a_1 a_4}$$

$$- \frac{a_1}{a_4} \cos(\phi_2) \tag{2.5}$$

And an expression for angle ϕ_3, as a function of ϕ_2 and ϕ_4, is obtained from the original loop component equations as follows:

$$\phi_3 = \tan^{-1}\left(\frac{a_2 \sin(\phi_2) - a_4 \sin(\phi_4)}{a_2 \cos(\phi_2) - a_4 \cos(\phi_4) - a_1}\right) \tag{2.6}$$

The plus and minus signs in equation denote two different angles which correspond to two possible linkage configurations as shown in Figure (2). For the slay mechanism the minus sign is taken.

The angular velocities $\dot{\phi}_3$ and $\dot{\phi}_4$ are found, for a given input crankshaft angular velocity $\dot{\phi}_2$ and linkage position ϕ_2, by differentiating displacement equations and complexity is reduced by various trigonometric identities, resulting in:

$$\dot{\phi}_3 = \frac{a_2 \sin(\phi_2 - \phi_4)\,\dot{\phi}_2}{a_3 \sin(\mu)} \tag{2.7}$$

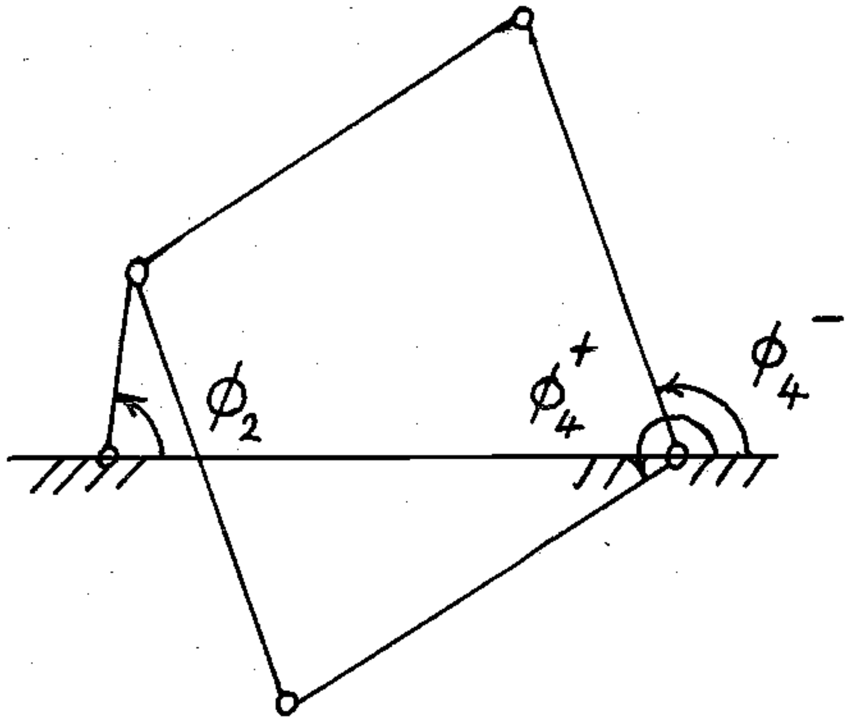


Figure 2. The two possible linkage configurations.

$$\dot{\phi}_4 = \frac{a_2 \sin(\phi_2 - \phi_3)\ \dot{\phi}_2}{a_4 \sin(\mu)} \tag{2.8}$$

Further, by differentiating and manipulating equations (2.7) and (2.8), equations for link angular accelerations $\ddot{\phi}_3$ and $\ddot{\phi}_4$ as functions of input crankshaft angular acceleration $\ddot{\phi}_2$, as well as velocity $\dot{\phi}_2$ and linkage position ϕ_2, are obtained:

$$\ddot{\phi}_3 = \frac{\dot{\phi}_3\,\ddot{\phi}_2}{\dot{\phi}_2} + \frac{a_1 a_2}{a_3^2}\left(\frac{\cos(\phi_2-\phi_4)\sin(\phi_4)\dot{\phi}_2}{\sin^2(\mu)} - \frac{\sin\phi_2\,\dot{\phi}_4}{}\right)\dot{\phi}_2$$

$$\ddot{\phi}_4 = \frac{\dot{\phi}_4\,\ddot{\phi}_2}{\dot{\phi}_2} + \frac{a_2 \cos(\phi_2-\phi_4)}{a_3 \sin(\mu)}\left(\dot{\phi}_2-\dot{\phi}_4\right)^2 + \frac{a_1}{a_3 a_4}\left(\frac{a_2\cos\phi_2\,\dot{\phi}_2^2 - a_4\cos(\phi_4)\dot{\phi}_4^2}{\sin(\mu)}\right)$$

As measured from origin 0 (which coincides with pivot A_1) the cartesian coordinates of the positions of each link centre of mass are given by:

$$x_2 = \xi_2$$

$$y_2 = n_2 \tag{2.11}$$

$$x_3 = \xi_3 + a_2\cos(\phi_2)$$

$$y_3 = n_3 + a_2\sin(\phi_2) \tag{2.12}$$

$$x_4 = \xi_4 + a_1$$

$$y_4 = n_4 \tag{2.13}$$

Upon differentiating the displacement equations, the center of mass velocity equations are obtained:

$$\dot{x}_2 = \dot{\xi}_2 = -\eta_2 \dot{\phi}_2$$
$$\dot{y}_2 = \dot{\eta}_2 = \xi_2 \dot{\phi}_2 \qquad (2.14)$$
$$\dot{x}_3 = -\eta_3 \dot{\phi}_3 - a_2 \sin(\phi_2)\dot{\phi}_2$$
$$\dot{y}_3 = \xi_3 \dot{\phi}_3 + a_2 \cos(\phi_2)\dot{\phi}_2 \qquad (2.15)$$
$$\dot{x}_4 = \dot{\xi}_4 = -\eta_4 \dot{\phi}_4$$
$$\dot{y}_4 = \dot{\eta}_4 = \xi_4 \dot{\phi}_4 \qquad (2.16)$$

Further differentiation gives the acceleration equations:

$$\ddot{x}_2 = \ddot{\xi}_2 = -\eta_2 \ddot{\phi}_2 - \xi_2 \dot{\phi}_2^2$$
$$\ddot{y}_2 = \ddot{\eta}_2 = \xi_2 \ddot{\phi}_2 - \eta_2 \dot{\phi}_2^2 \qquad (2.17)$$
$$\ddot{x}_3 = -\eta_3 \ddot{\phi}_3 - \xi_3 \dot{\phi}_3^2$$
$$\quad - a_2 (\sin(\phi_2)\ddot{\phi}_2 + \cos(\phi_2)\dot{\phi}_2^2)$$
$$\ddot{y}_3 = \xi_3 \ddot{\phi}_3 - \eta_3 \dot{\phi}_3^2$$
$$\quad + a_2 (\cos(\phi_2)\ddot{\phi}_2 - \sin(\phi_2)\dot{\phi}_2^2 \qquad (2.18)$$
$$\ddot{x}_4 = \ddot{\xi}_4 = -\eta_4 \ddot{\phi}_4 - \xi_4 \dot{\phi}_4^2$$
$$\ddot{y}_4 = \ddot{\eta}_4 = \xi_4 \ddot{\phi}_4 - \eta_4 \dot{\phi}_4^2 \qquad (2.19)$$

And the equations for the x and y components of the bearing reaction forces, crankshaft input moment M_{IN} and shaking force and moment transmitted to the loom frame as follows:

Bearing Reaction Forces

$$F_{12x} = m_2 \ddot{x}_2 + m_3 \ddot{x}_3 + D_3 \cos(\phi_4) + D_4 \cos(\phi_3)$$

$$F_{12Y} = m_2 \ddot{y}_2 + m_3 \ddot{y}_3 + D_3 \sin \phi_4 + D_4 \sin \phi_3 \quad (2.20)$$

$$F_{32X} = -m_3 \ddot{x}_3 - D_3 \cos \phi_4 - D_4 \cos \phi_3$$

$$F_{32Y} = -m_3 \ddot{y}_3 - D_3 \sin \phi_4 - D_4 \sin \phi_3 \quad (2.21)$$

$$F_{34X} = D_3 \cos \phi_4 + D_4 \cos \phi_3$$

$$F_{34Y} = D_3 \sin \phi_4 + D_4 \sin \phi_3 \quad (2.22)$$

$$F_{14X} = m_4 \ddot{x}_4 - D_3 \cos \phi_4 - D_4 \cos \phi_3$$

$$F_{14Y} = m_4 \ddot{y}_4 - D_3 \sin \phi_4 - D_4 \sin \phi_3 \quad (2.23)$$

Crankshaft Input Moment

$$M_{IN} = \frac{1}{\dot{\phi}_2} \sum_{i=2}^{4} m_i (\dot{x}_i \ddot{x}_i + \dot{y}_i \ddot{y}_i + k_i^2 \dot{\phi}_i \ddot{\phi}_i) \quad (2.24)$$

Shaking Forces

$$F_{SX} = -m_2 \ddot{x}_2 - m_3 \ddot{x}_3 - m_4 \ddot{x}_4$$

$$F_{SY} = -m_2 \ddot{y}_2 - m_3 \ddot{y}_3 - m_4 \ddot{y}_4 \quad (2.25)$$

Shaking Moment

$$M_{SO} = -\sum_{i=2}^{4} m_i (x_i \ddot{y}_i - y_i \ddot{x}_i + k_i^2 \ddot{\phi}_i) \quad (2.26)$$

where:

$$D_i = \frac{m_i}{a_i \sin(\mu)} \left(\eta_i \ddot{x}_i - \xi_i \ddot{y}_i - k_i^2 \ddot{\phi}_i \right)$$

$$(i = 3, 4)$$

and K_i is the radius of gyration of link i with respect to link centre of mass. All kinematic parameters and dynamic quantities are discrete, and have to be calculated for all positions of the linkage, i.e., $0 < \phi_2 < 2\pi$.

For the slay mechanism, computer programs were developed to compute the quantities mentioned above for all positions of the linkage using a discrete step size of 2 degress of crankshaft rotation angle ϕ_2. For the slay, $\ddot{\phi}_2$ has been taken to be zero i.e. no variation of crankshaft speed within a weaving cycle in order to simplify the analysis. Friction losses in the bearings can be approximately evaluated by computing the total force on any bearing, given by:

$$F_{Ai} = \sqrt{F_{AiX}^2 + F_{AiY}^2} \qquad (2.27)$$

where $F_{A1X} = F_{12X}$, $F_{A2X} = F_{32X}$, $F_{A3X} = F_{34X}$, $F_{A4X} = F_{14X}$ and similarly for F_{A1Y}, F_{A2Y}, F_{A3Y}, and F_{A4Y}. Then the power required at any instant to overcome a bearing friction will be:

$$P_{Ai} = F_{Ai} R_{Ai} \mu_{Ai} \dot{\phi}_{Ai} C \qquad (2.28)$$

where:

P_{Ai} = power required at any bearing A_i

R_{Ai} = radius of bearing A_i

$_{Ai}$ = coefficient of friction for bearing A_i

$\dot{\phi}_{Ai}$ = absolute value of relative angular rotation of the two links attached to joint A_i

C = numerical constant

By adding up all the components of power required to overcome bearing friction losses and dividing by $\dot{\phi}_2$, an approximate value for friction torque M_f as it is reflected on the crankshaft and hence on the loom motor can be obtained.

The average and root mean square values have been evaluated for such important parameters as ϕ_4, $\dot{\phi}_4$, $\ddot{\phi}_4$, M_{IN}, M_f, M_{SO}, F_S, F_{A1}, F_{A2}, F_{A3} and F_{A4}. Table (2) lists the various kinematic and dynamic parameters of the Draper X - 2 loom used. Plots of such quantities as ϕ_4, $\dot{\phi}_4$, $\ddot{\phi}_4$, M_{IN}, M_f, M_{SO}, F_S, F_{A1}, F_{A2}, F_{A3}, and F_{A4} versus ϕ_2 for $\dot{\phi}_2$ = 20.95 radians/second (200 picks per minute) are shown in Figures (3) to (13). In the case of friction torque, μ_{Ai} (i=1 to 4) has been taken to be 0.15. This is the maximum value of friction for steel shafts running in grease lubricated bearings. The average and R.M.S. values of these quantities as well as friction power and other related quantities are listed in Table (3).

Shedding Mechanism

Although many variations exist, for the purpose of this thesis the most basic type of shedding motion has been studied. This mechanism is a cam operated system with a treadle lever which raises and lowers the harness, against the action of a spring. In the simplest case there are two harnesses and thus two cams. Figures (14 and 15) show the arrangement in a simplified form. The following nomenclature

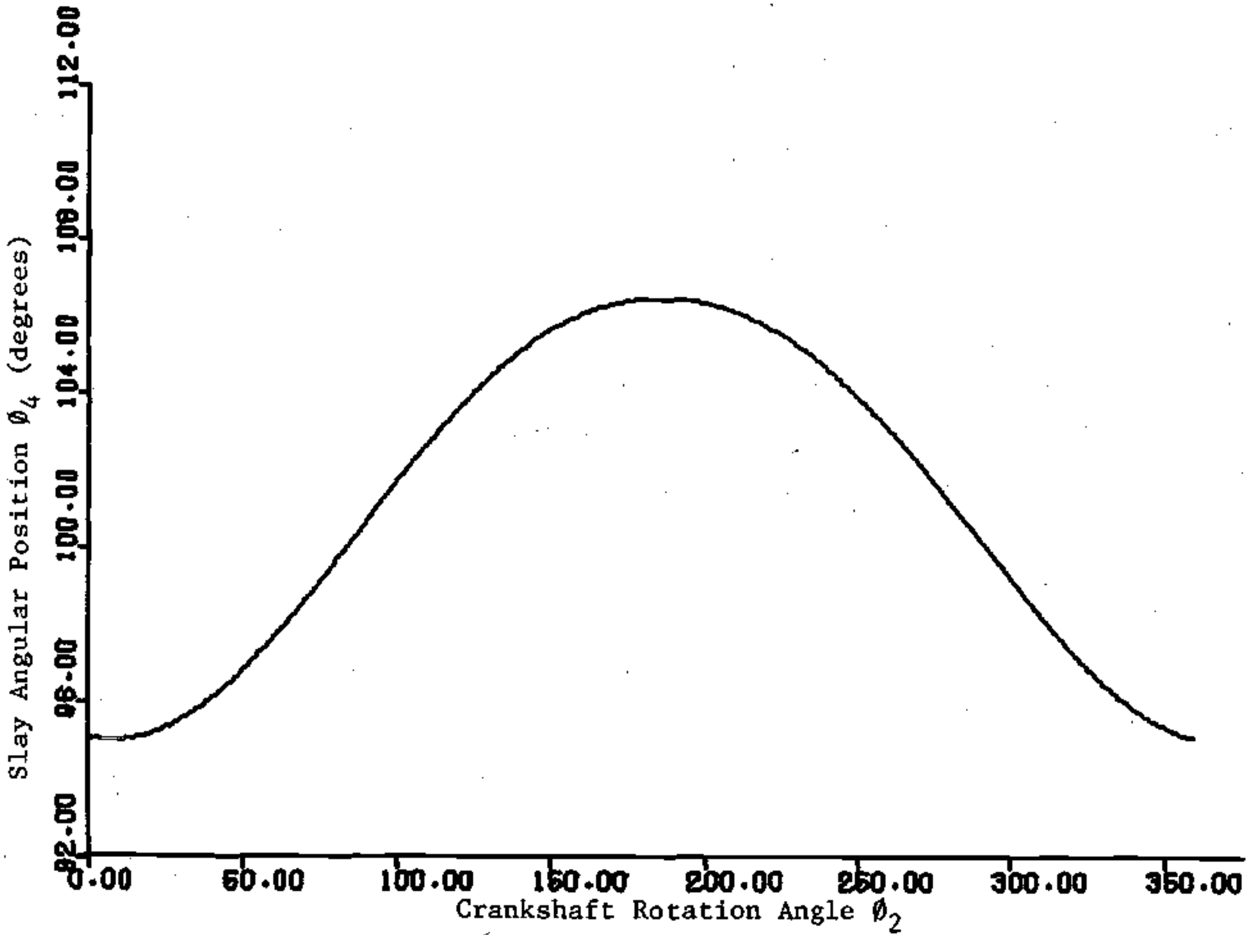


Figure 3. Slay Angular Position $\emptyset_4$ Versus Crankshaft Rotation Angle $\emptyset_2$ (degrees) (for Draper X - 2 Loom)

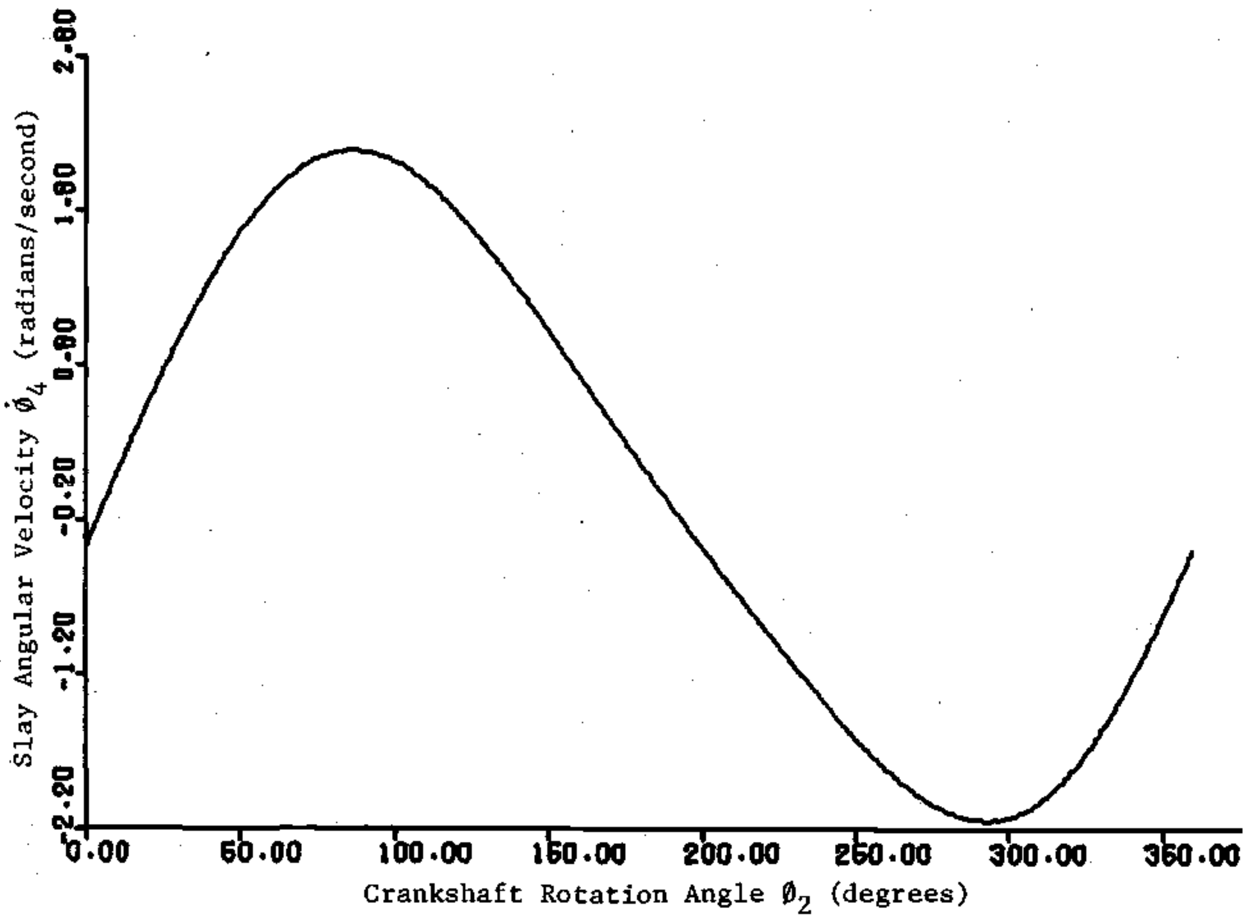


Figure 4. Slay Angular Velocity $\dot{\phi}_4$ (radians/sec) versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/sec (200 p.p.m.)] (For Draper X - 2 Loom)

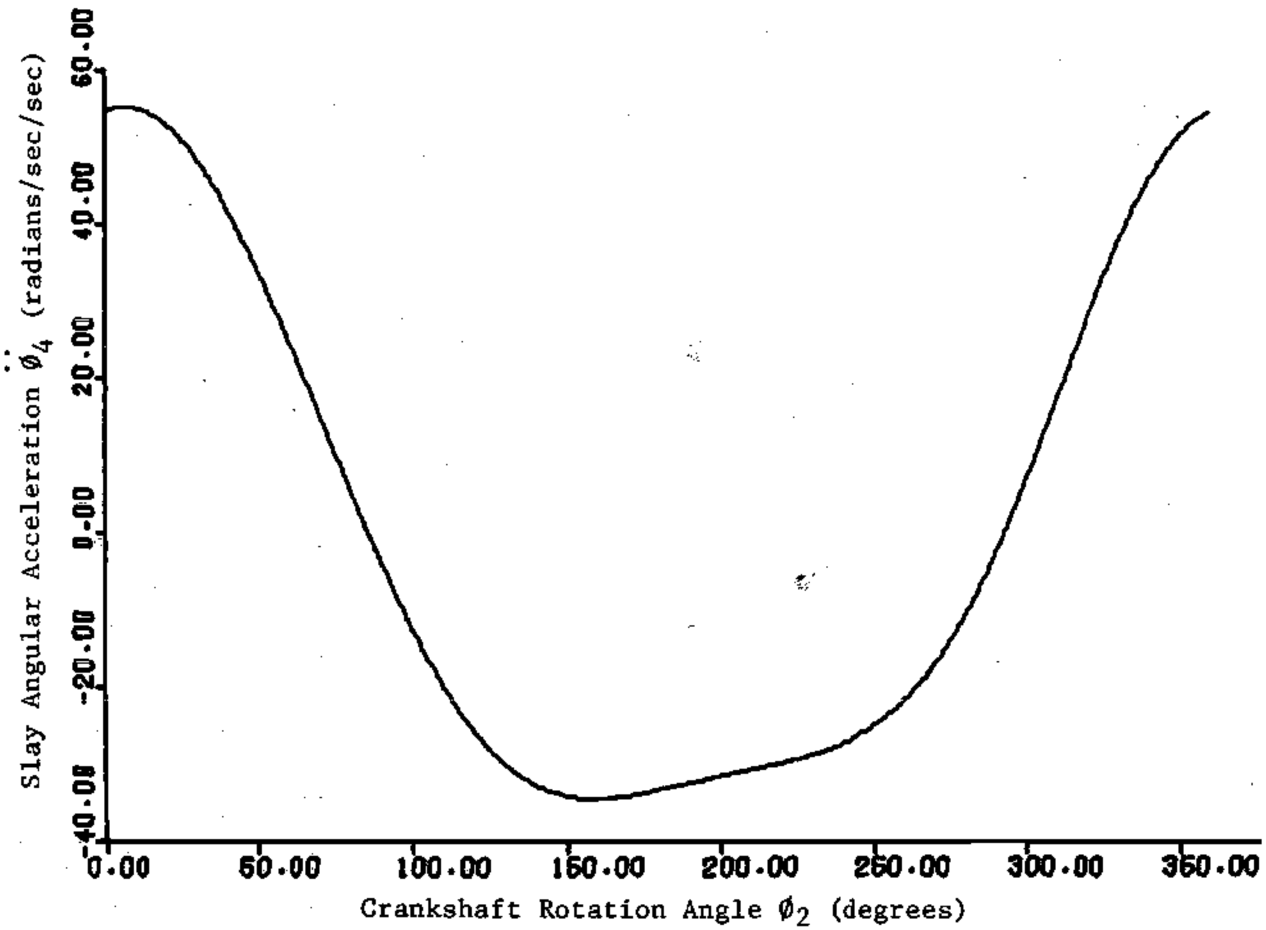


Figure 5. Slay Angular Acceleration $\ddot{\phi}_4$ (radians/second/second) versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/ p.p.m.), $\ddot{\phi}_2$ = 0.0 radians/sec/sec] (For Draper X - 2 Loom)

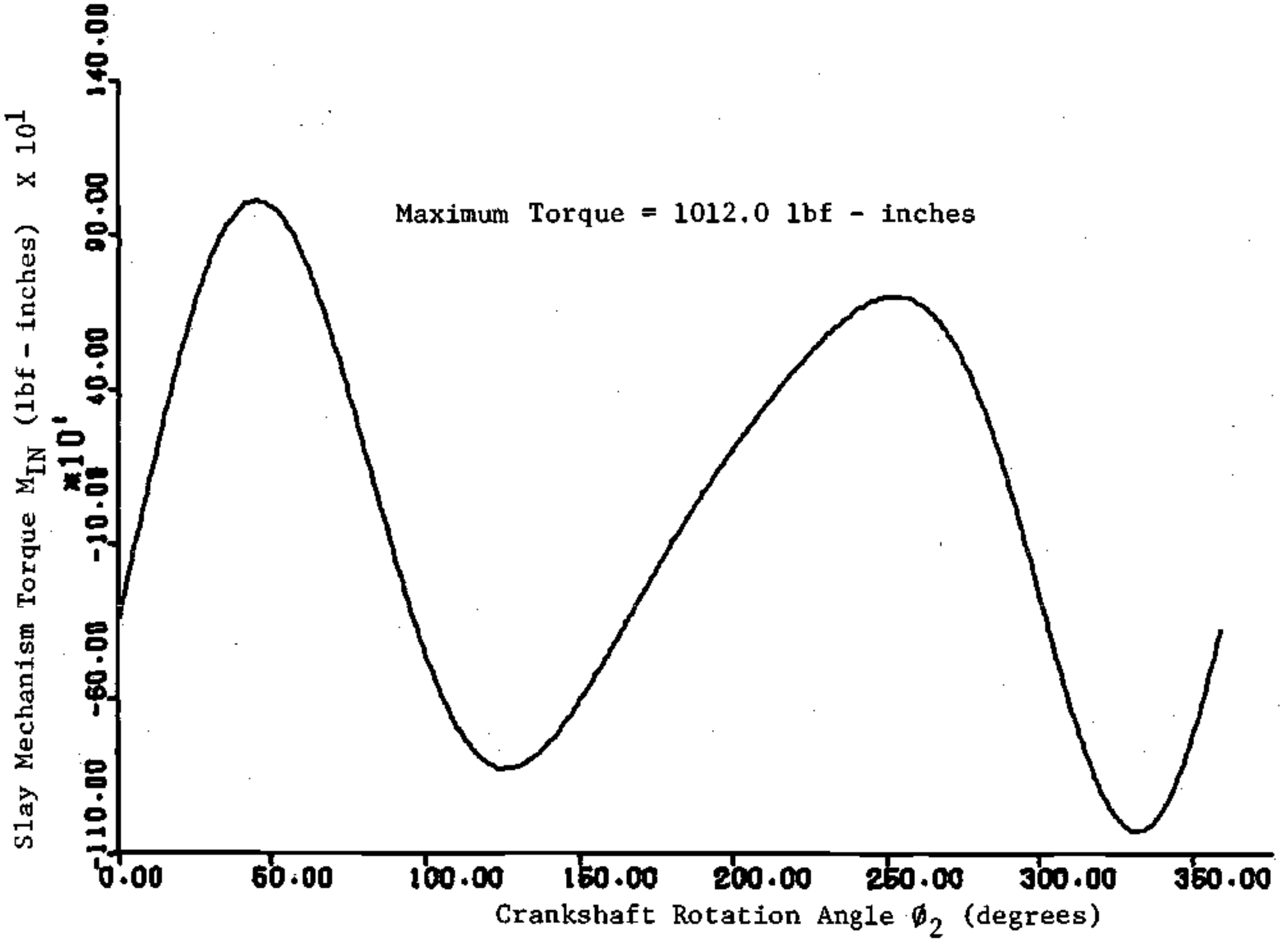


Figure 6. Required Slay Mechanism Input Torque (M_{IN}) (lbf - inches) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second/ 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

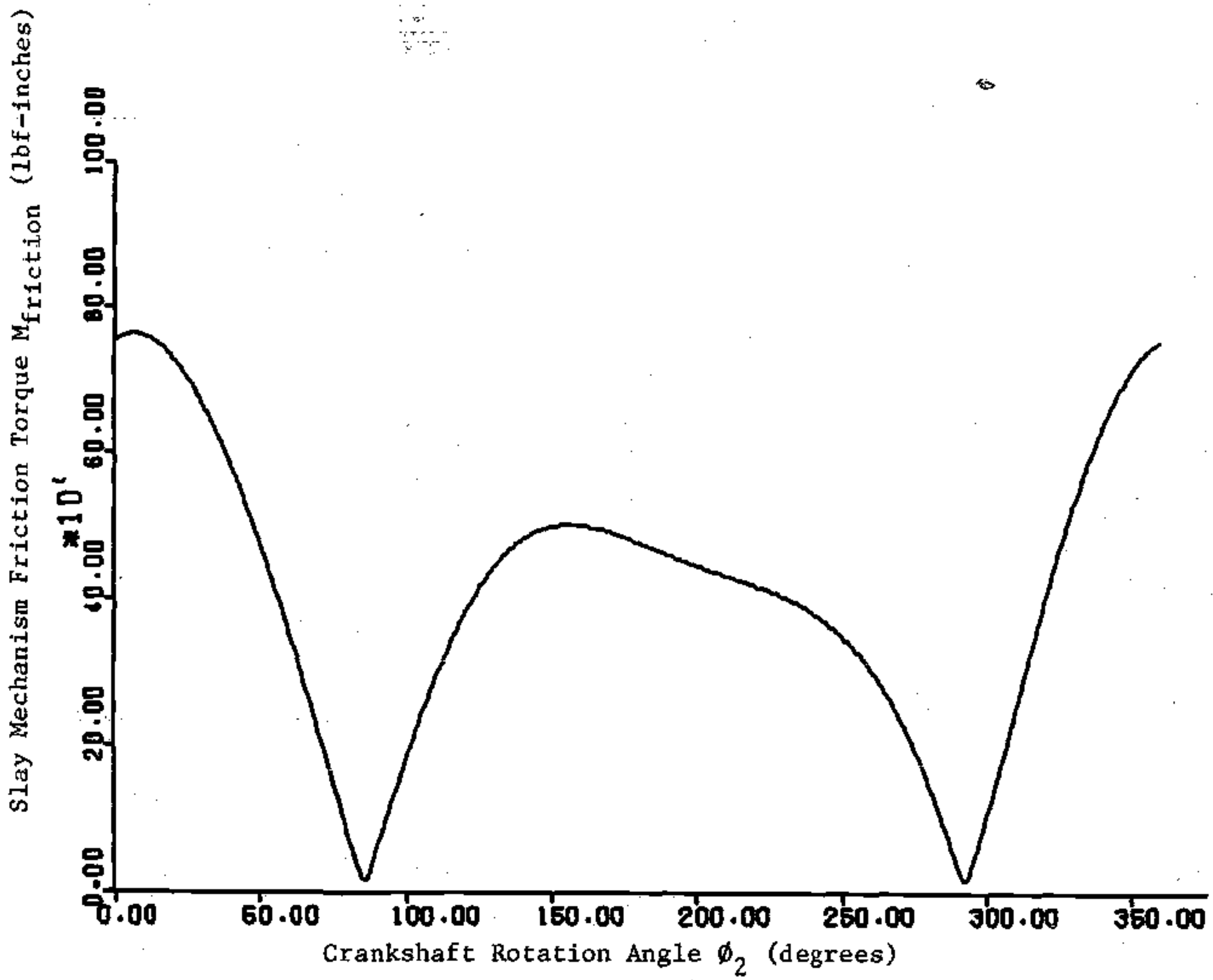


Figure 7. Slay Mechanism Friction Torque M_f (lbf - inches) Versus Crankshaft Rotation Angle ϕ_2 (degrees). [$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m, $\ddot{\phi}_2$ = 0.0 radians/second/second, μ = 0.15] (For Draper X - 2 Loom)

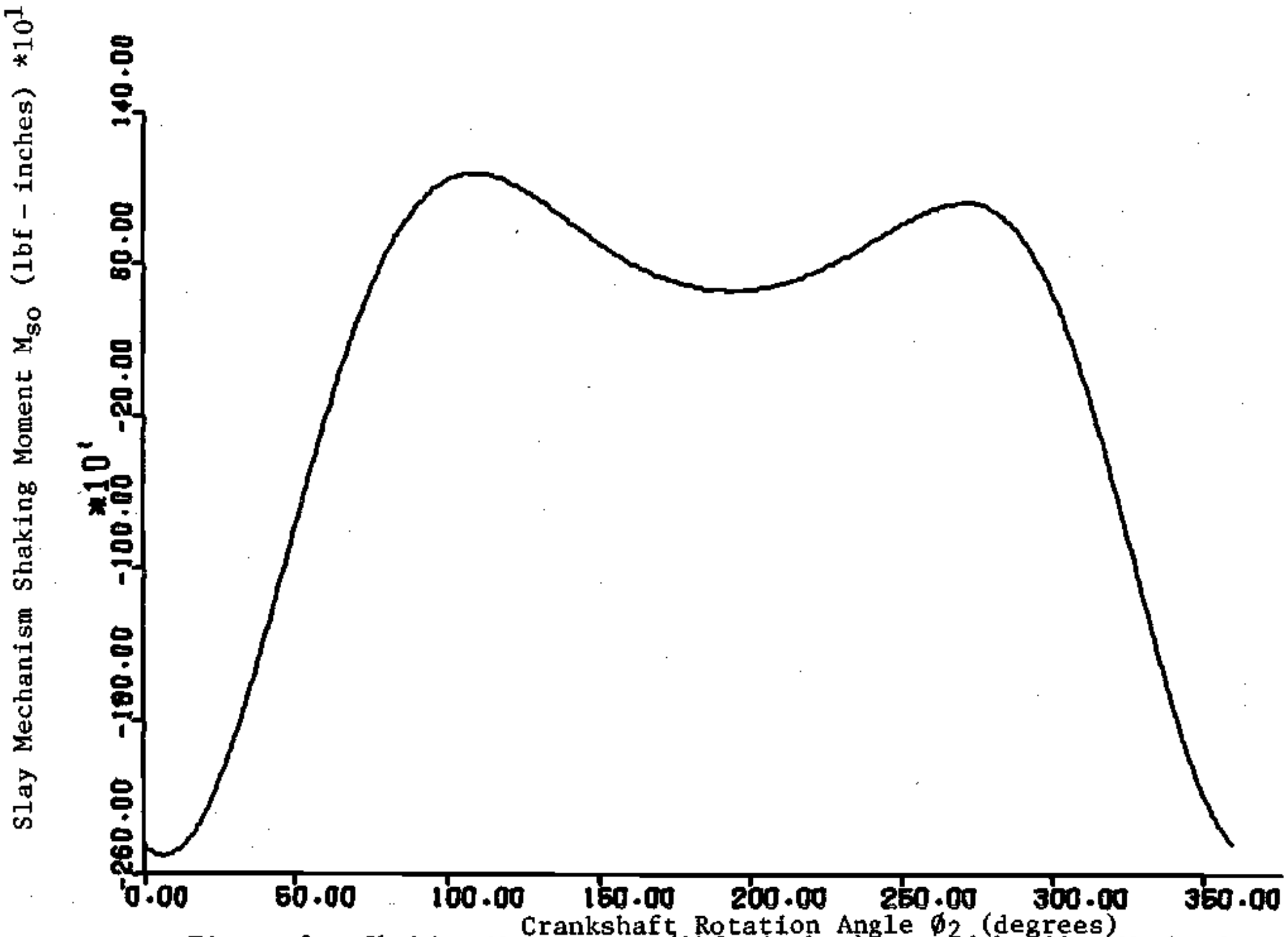


Figure 8. Shaking Moment M_{so} (lbf - inches) caused by Slay Mechanism and Transmitted to Loom-Frame Versus Crankshaft Rotation Angle $\emptyset_2$ (degrees) [$\dot{\emptyset}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\emptyset}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

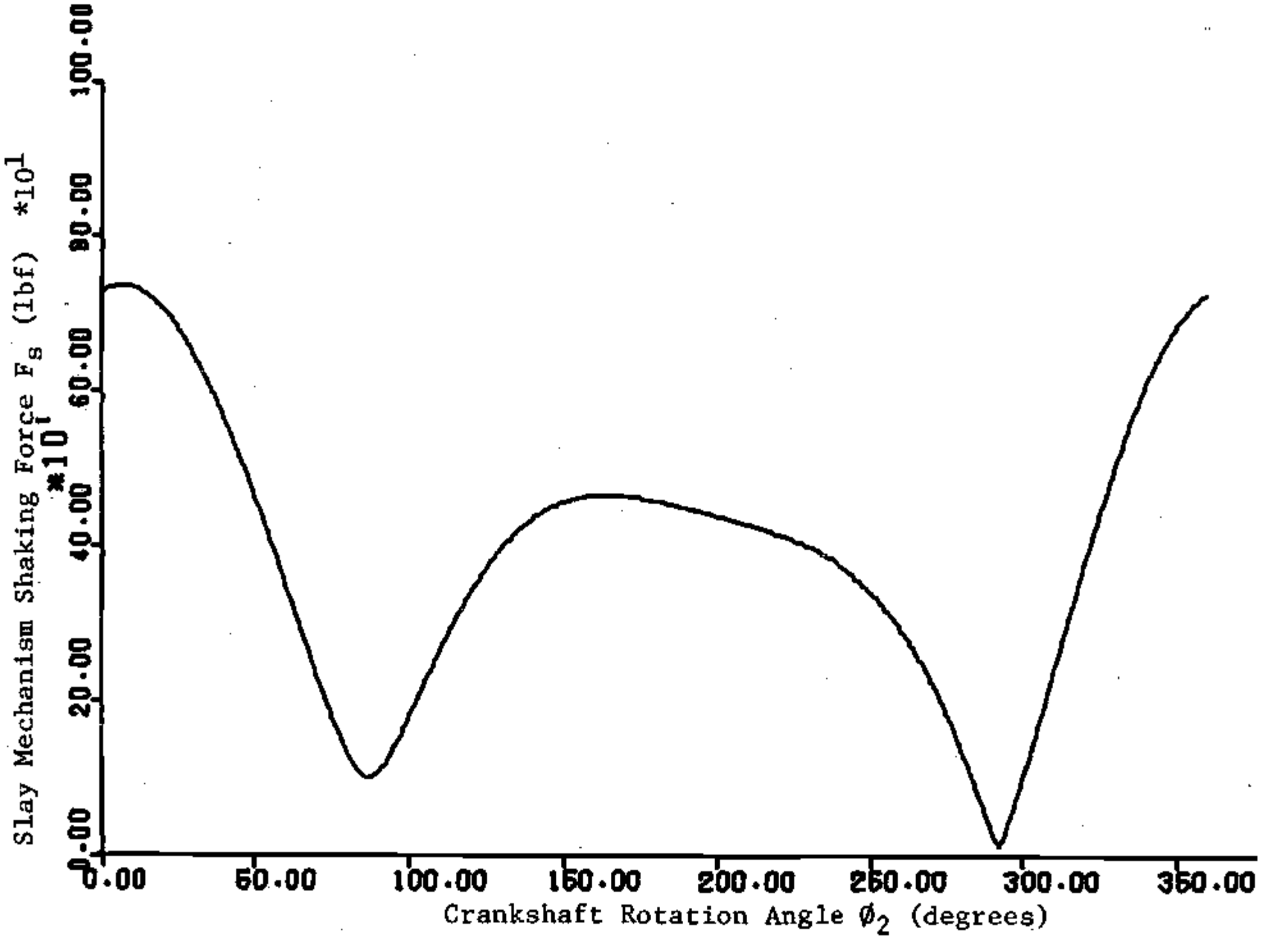


Figure 9. Shaking Force F_s (lbf) caused by Slay Mechanism and Transmitted to Loom Frame Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second - 200 p.p.m. , $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

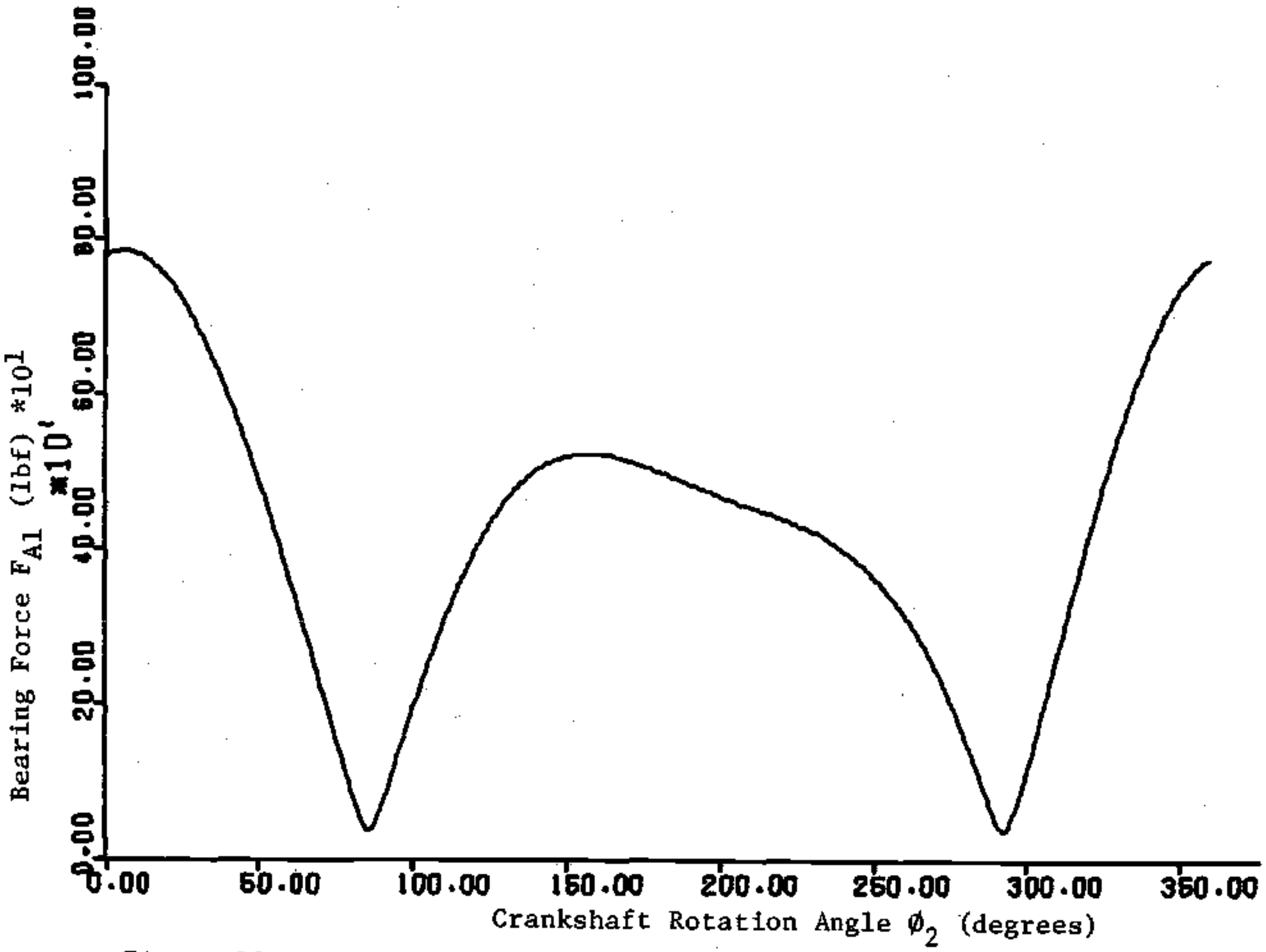


Figure 10. Bearing Force F_{A1} (lbf) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

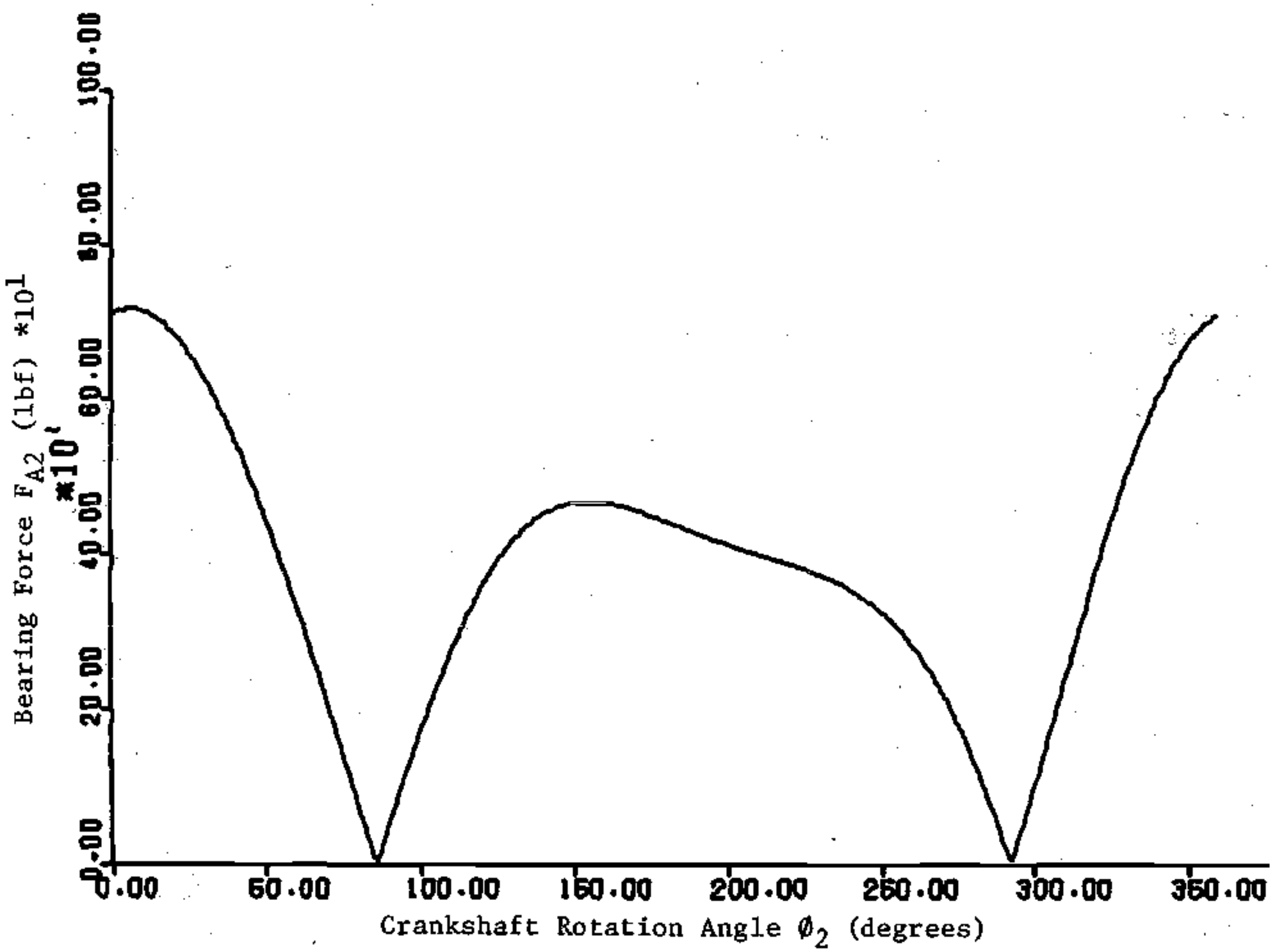


Figure 11. Bearing Force F_{A2} (lbf) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second - 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

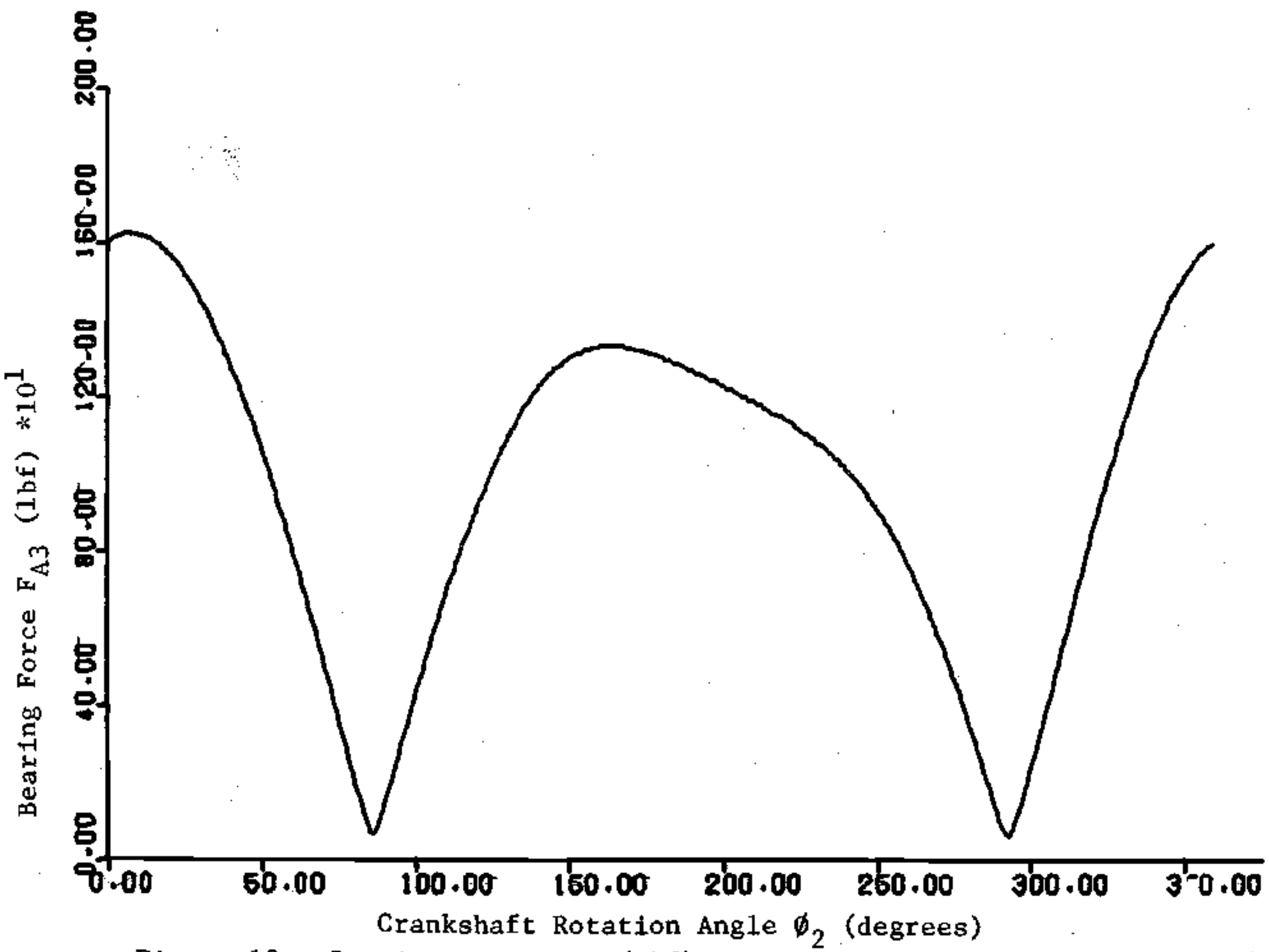


Figure 12. Bearing Force F_{A3} (lbf) Versus Crankshaft Rotation Angle ϕ_2 (degrees)
[$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second]
(For Draper X - 2 Loom)

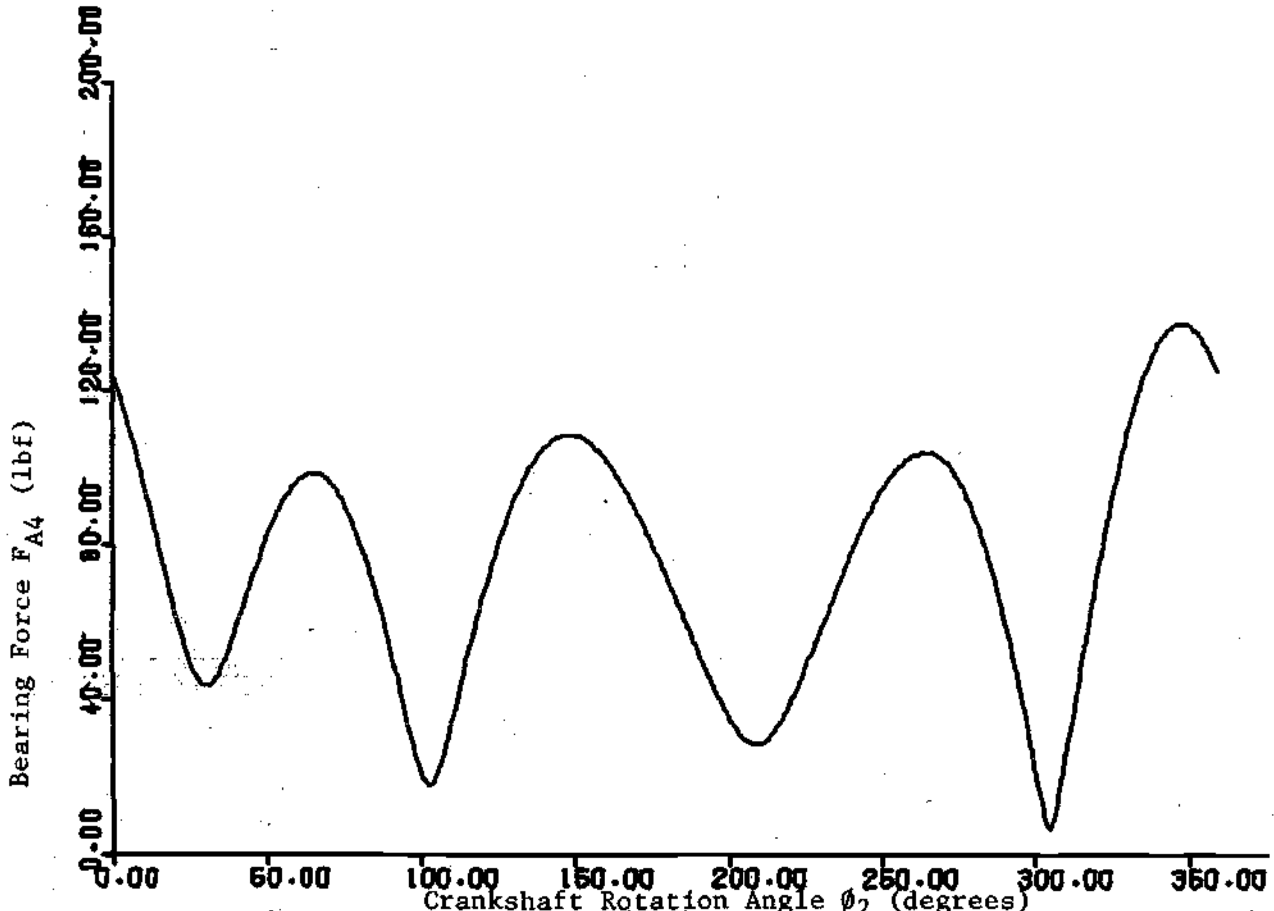


Figure 13. Bearing Force F_{A4} (lbf) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second]

Table 3. Average and R.M.S. Values of Various Kinematic and Dynamic Parameters for the Slay Mechanism (Draper X - 2 Loom)

Parameter	Average Value	R.M.S. Value
Slay Velocity ($\dot{\phi}_4$) (radians/second)	0.0	1.496
Slay Acceleration ($\dot{\phi}_4$) (radians/second/second)	0.63	32.55
Slay torque (lbf-inches)	0.0	616.12
Slay friction torque (lbt-inches)	107.46 (μ=0.15)	118.67 (μ=0.15)
Shaking force (lbf)	402.9	441.35
Shaking Moment (lbf-inches)	-27.2	1161.5
Nominal Power (Watts)	0.0	1458.57
Friction Power (Watts)	253.8	280.93
Bearing force F_{12} (lbf)	427.7	615.9
Bearing force F_{32} (lbf)	411.50	573.2
Bearing force F_{34} (lbf)	384.5	503.1
Bearing force F_{14} (lbf)	76.0	187.3

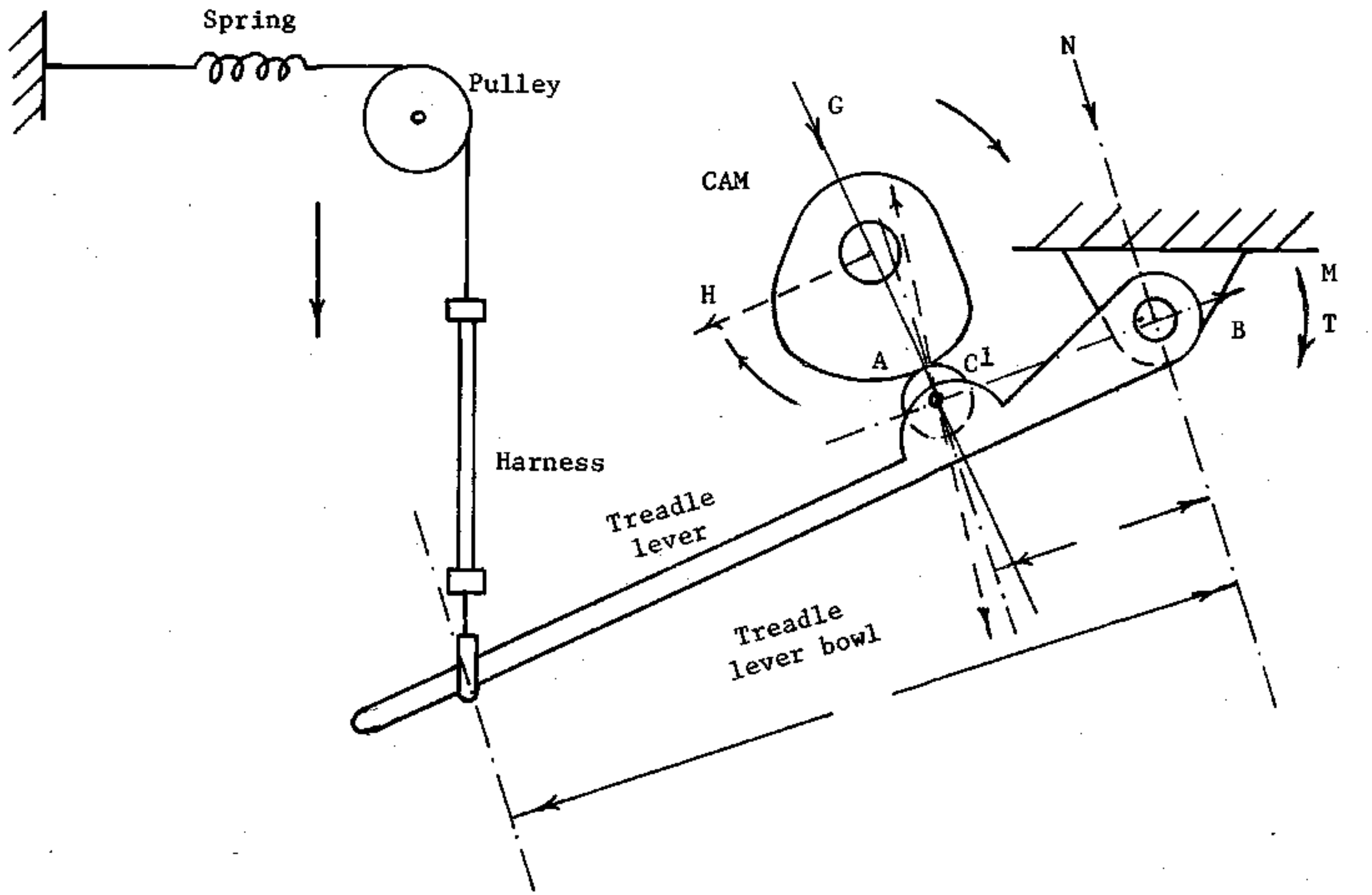


Figure 14. Shedding Cam Mechanism

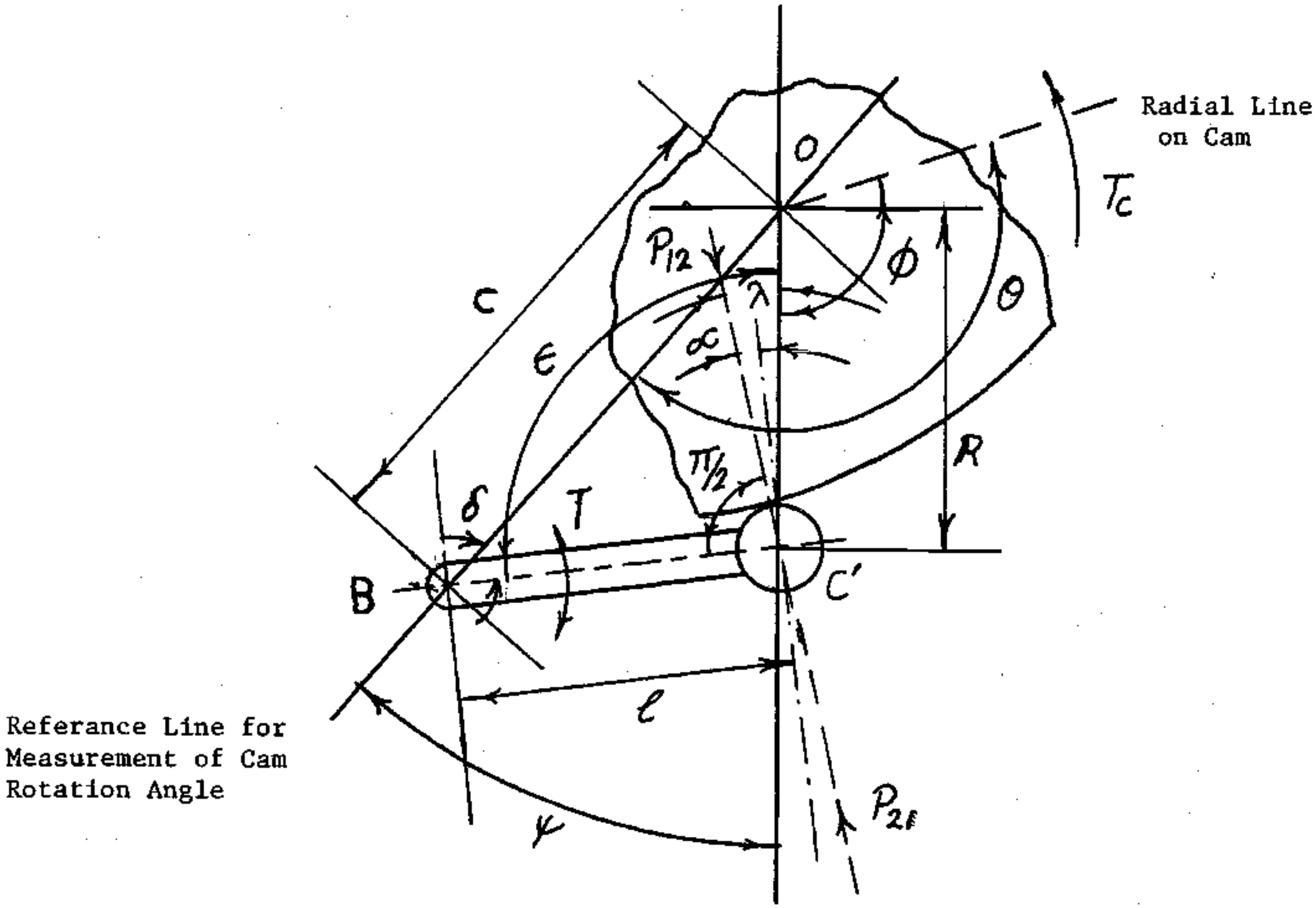


Figure 15. Magnified View of Cam - Cam Follower Area Shown in Figure (14).

is used for the purpose of shedding motion analysis:

- A = point of contact of treadle lever bowl with cam
- B = position of treadle lever pivot
- c = distance from cam centre to treadle lever pivot
- C^1 = position of treadle lever bowl center
- 1 = distance between treadle lever pivot and treadle lever bowl center
- P_{12} = normal force acting on treadle lever bowl
- O = cam center
- R = distance between cam shaft center and treadle lever bowl center
- R_o = minimum distance between camshaft center and treadle lever bowl center
- T = torque acting on treadle lever
- α = pressure angle at any point
- β = total angular rotation of cam for a complete shedding cycle (a rise or a fall)
- δ = angle OBC
- ϵ = Angle BCO
- θ = angular rotation of cam measured from a space fixed referance line
- λ = angle between normal to cam surface and the line OC
- τ = total angular displacement of treadle lever for a complete shedding cycle
- ϕ = angular rotation of cam measured from a line OC joining cam and treadle lever bowl centers
- ψ = Angle C OB

P_{12} = reaction of the cam

P_{21} = $-P_{12}$

M = reaction force at bearing center B acting along line BC

N = reaction force at bearing center B acting perpendicular to line BC

G = reaction at camshaft bearing acting on cam along line OC

H = reaction at camshaft bearings acting on cam along a line perpendicular to OC

k = spring constant

χ_o = effective pressure angle which includes treadle lever bowl bearing friction effects

χ_{10} = basic pressure angle which includes camshaft bearing friction effects

μ_o = effective friction coefficient of roller bearing at treadle lever bowl center

μ = coefficient of friction at camshaft bearings

L = effective length of treadle levers

T_c = camshaft torque

R_c = radins of camshaft

The kinematic and dynamic equations for a similar cam-follower mechanism have been developed by Kloomok and Muffley[9]. Angle δ is a function of angle θ. For the shedding mechanism, δ has a simple harmonic motion with respect to θ, which may be represented by:

$$\delta = \delta_o + \frac{\zeta}{2}\left[1 - \cos\left(\frac{\pi\theta}{\beta}\right)\right] \qquad (2.29)$$

Angle δ is similar to pressure angle, and can be represented by the following equation:

$$\lambda = \tan^{-1} \frac{1}{R} \frac{dR}{d\theta} \tag{2.30}$$

Once δ is know, the triangle OBC^1 for any position in Figure (15) may be solved for R, ϵ and ψ.

$$R = \sqrt{\ell^2 + c^2 - 2\ell c \cos(\delta)} \tag{2.31}$$

$$\epsilon = \sin^{-1}\left[\frac{c}{R} \sin(\delta)\right] \tag{2.32}$$

$$\psi = \cos^{-1}\left(\frac{c^2 + R^2 - \ell^2}{2Rc}\right) \tag{2.33}$$

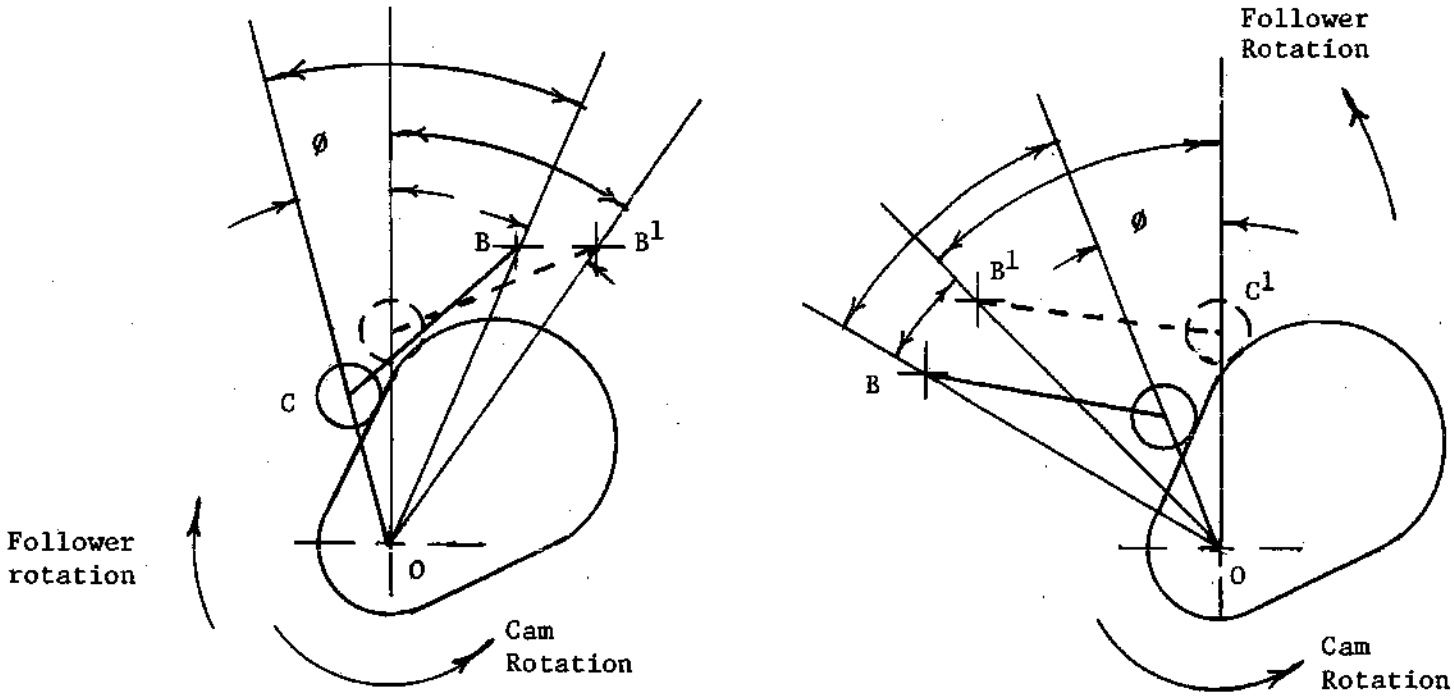
For cams actuating swinging followers, there are two possible types of relative location and rotation of the winging follower with respect to cam rotation as shown in Figure 916). Figure (16a) shows one type in which the cam rotates "away from" the pivot and figure (16b) shows the other type in which the cam rotates "towards" the pivot. For the shedding mechanism, the cam rotates "towards" the pivot. As can be seen from Figure (15):

$$\alpha = -\frac{\pi}{2} + \epsilon + \lambda \tag{2.34}$$

$$\phi = \psi - (\psi_0 - \theta) \tag{2.35}$$

Angle ϵ as measured from BC^1 in a clockwise direction is positive and angle λ as measured from line OC^1 in a counterclockwise direction is negative.

Differentiating equation (2.35) with respect to R:



(a)"from" pivot

(b) toward pivot

Figure 16. Swinging Follower Rotation "from" and "toward" the pivot.

$$\frac{d\phi}{dR} = \frac{d\theta}{dR} + \frac{d\psi}{dR} \qquad (2.36)$$

Differentiating equation (2.31) with respect to θ and taking the reciprocal:

$$\frac{d\theta}{dR} = \frac{R}{\ell c \sin\delta \left(\frac{d\delta}{d\theta}\right)} \qquad (2.37)$$

Differentiating equation (2.34) with respect to R:

$$\frac{d\psi}{dR} = \frac{c^2 - R^2 - \ell^2}{2R^2 c \sin\psi} \qquad (2.38)$$

Collecting terms from equations (2.30), (2.32), (2.37), (2.38) and (2.36) and substituting in equation (2.34):

$$\alpha = -\frac{\pi}{2} + \sin^{-1}\left[\frac{c}{R}\sin\delta\right]$$

$$+ \tan^{-1}\left[\frac{1}{\dfrac{R^2}{\ell c \sin\delta \frac{d\delta}{d\theta}} + \dfrac{c^2 - R^2 - \ell^2}{2Rc \sin\psi}}\right] \qquad (2.39)$$

This completes the derivation of the kinematic equations for shedding motion.

The dynamic equations for shedding motion have been developed in a manner similar to that followed by Kloomok and Muffley. As shown in Figure (14), the torque T acting on treadle lever due to spring pressure (excluding inertia effects of the treadle lever and harnesses) will be:

$$T = kL^2 \cos\delta \sin\delta + \text{CONSTANT} \tag{2.40}$$

The constant is for the torque due to preloading of the spring. Force P_{21} and Reactions M and N are in equilibrium and thus satisfy the conditions:

$$\sin(\chi_o) P_{21} - M = 0$$

$$\cos(\chi_o) P_{21} - N = 0 \tag{2.41}$$

The moments about pivot 0 are due to force P_{21}, namely $\ell \cos (N_o) P_{21}$ and the load torque T. The friction torque due to bearing friction has been included by using angle χ_o which includes the effect of coefficient of friction μ_o. The friction torque due to reactions at bearing 0, has been neglected because it is very small. Thus neglecting the inertia effects the equation for equilibrium about point 0 is:

$$\ell \cos(\chi_o) P_{21} = T \tag{2.42}$$

Equations (2.41) and (2.42) yield the following solutions:

$$P_{21} = \frac{T}{\ell} \sec(\chi_o) \tag{2.43}$$

$$M = \frac{T}{\ell} \tan(\chi_o) \tag{2.44}$$

$$N = \frac{T}{\ell} \tag{2.45}$$

Then the camshaft torque and bearing reactions can be evaluated by the following equations:

$$G = \cos(\chi_{10}) P_{12} \qquad (2.46)$$

$$H = \sin(\chi_{10}) P_{12} \qquad (2.47)$$

$$T_c = P_{12} (R \sin(\chi_{10}) + \mu R_c) \qquad (2.48)$$

In deriving equation (2.48), the inertial effects due to speed fluctuations of camshaft have been neglected and the camshaft is assumed to rotate at a constant speed. Table (4) lists the values of the various shedding mechanism parameters for the X - 2 loom. The shedding torque, as it is reflected on the crnakshaft as a function of crankshaft rotation angle ϕ_2 is shown in Figure (17). The average power required per harness is 15.2 watts, at a loom speed of 200 picks per minute. For the simplest case of TWC harnesses the power requirement is 30.4 watts.

Picking Mechanism

The picking mechanism is rather complicated by design and has many flexible members made of wood and leather. Therefore a very elaborate treatment of the motion has been avoided. The inertia forces involved in picking, though considerable, have only been evaluated experimentally. The torque required to accelerate the shuttle has been evaluated using equations derived by Catlow[11]. The following nomenclature has been used:

ϕ = angle of rotation of camshaft (radians)

W = angular velocity of camshaft (radians per second)

S = nominal distance moved by the shuttle and picker (inches)

t = elasped time (seconds)

Table 4. Shedding Mechanism Details
(Draper X - 2 Loom)

Parameter	Numerical Value
c	10.0 inches
l	9.25 inches
B	90.0 degrees
τ	10.14 degrees
δ_o	17.0 degrees
k	48.25 lbf per inch
μ_o	0.0018
μ	0.15
L	20.29 inches

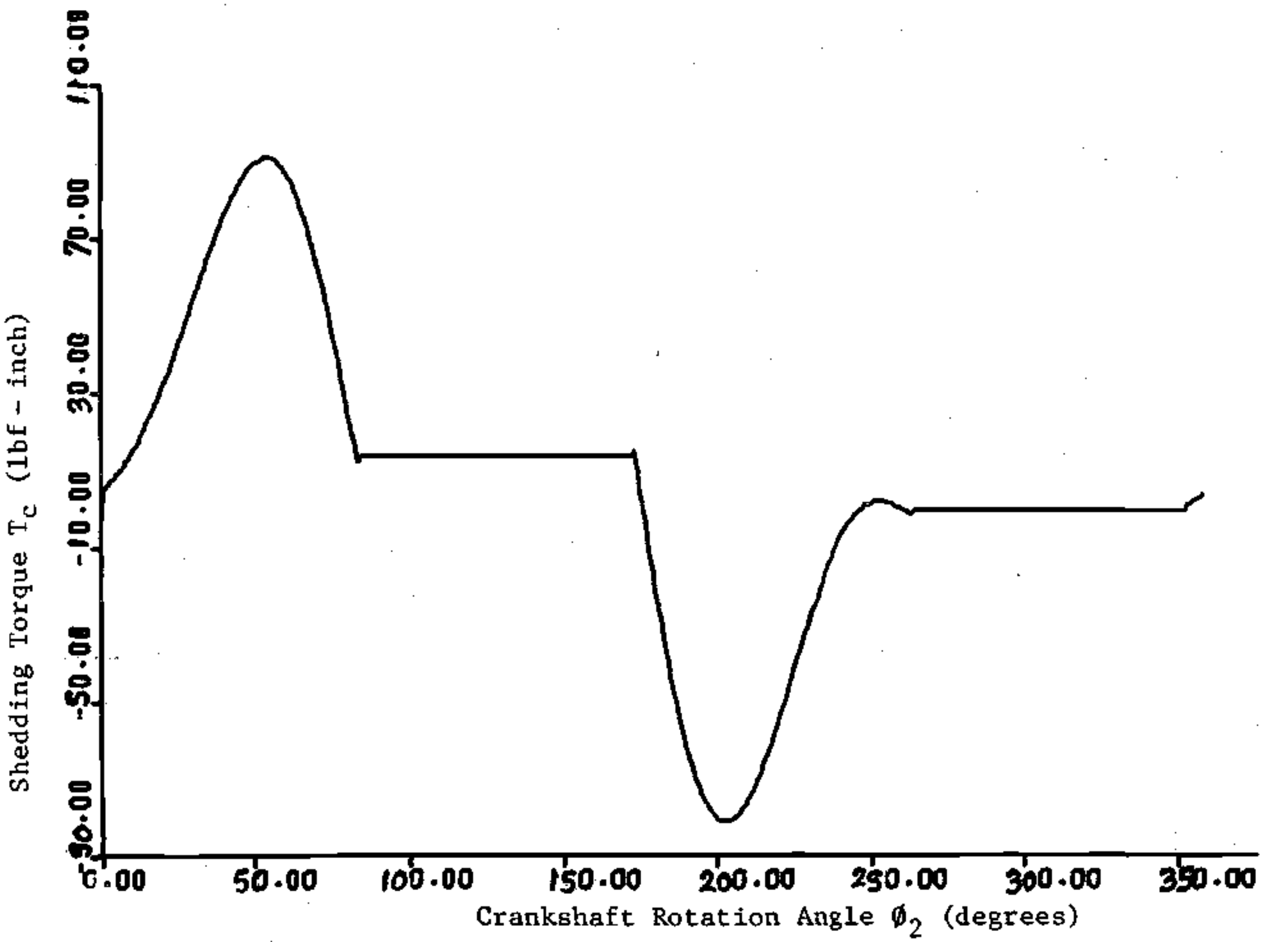


Figure 17. Shedding Torque T_c lb - inch (per harness) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X -2 Loom)

p = constant (units of length, i.e., inch)

M = mass of moving parts like shuttle, picker and upper third of picking stick (slugs)

n = alacrity of the system (second $^{-1}$)

x = actual distance moved by the shuttle and picker (inches)

λ = rigidity of the mechanism (pounds per foot)

F = force at the top of the picker stick (pounds)

T = camshaft torque (pound - inch)

C = constant (inch)

Assuming that the camshaft speed is constant the angle of rotation of the camshaft in time t will be:

$$\phi = \omega t \tag{2.49}$$

On most looms, S, the nominal distance (also called the designed distance) moved by the shuttle and picker in time t has a linear relationship with respect to the angular rotation of the camshaft:

$$S = p\phi \tag{2.50}$$

The fundamental equation of movement of the system is:

$$M \frac{d^2x}{dt^2} = \lambda (S - x) \tag{2.51}$$

The alacrity of the system is defined by the relation:

$$n^2 = \frac{\lambda}{M} \tag{2.52}$$

Thus equation (2.51) can be written in the form:

$$\frac{1}{n^2} \frac{d^2x}{dt^2} + x = S \tag{2.53}$$

When S from equation (2.53) is substituted in equation (2.50) and integrated, the force F at the top of the stick can be found by:

$$F = M p \omega n \sin(nt) \qquad (2.54)$$

Once F is known the torque T that the camshaft has to deliver can be approximately evaluated as:

$$T = FC \qquad (2.55)$$

In equation (2.55), C is a constant which has the units of length and includes the nominal multiplication factor ncessary to relate the force at the top of the picker stick to that which acts at the cam roller follower and the nominal torque arm of the displaced roller follower. After the torque has been evaluated the picking power can be easily estimated. Table (5) lists the various important parameters for the picking system of the Draper X - 2 loom. Figure (18) shows the picking torque as a function of crank angular position. The average power required for picking is 66.8 watts. The R.M.S. power required is 263.66 watts. This excludes the power required to accelerate the mechanism parts themselves and this has only been evaluated experimentally.

Total Driving Torque

The summation of the theoretically calculated torques required for picking beating up and shedding gives the total driving torque as required these three mechanisms at any time. Figure (19) shows the value of this torque for any crank rotation angle. This driving torque however does not take into account the inertia and friction losses in the picking mechanism as well as the torque required by the other loom mechanisms such as let off, take up, warp stop motion, etc.

Table 5. Picking Mechanism Parameters

(Draper X - 2 Loom)

Parameter	Numerical Value
w	10.45 radians/second
P	0.5 inch/degree
M	0.11 slugs
n	132.0 second^{-1}
λ	1916.64 slugs/second^2
C	9.71 inches

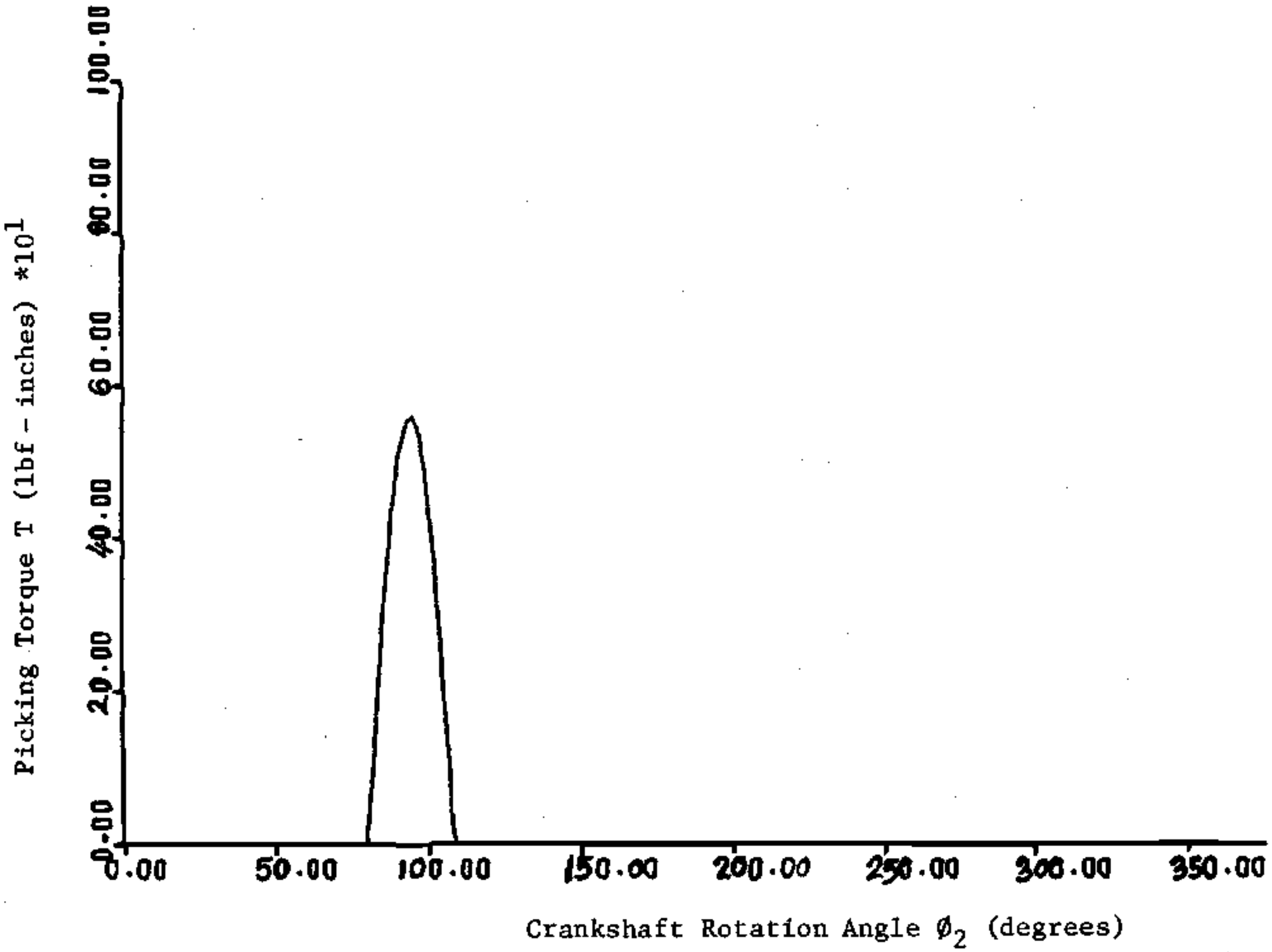


Figure 18. Required Picking Torque T (lbf – inches) as Reflected on the Crankshaft Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second – 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X – 2 Loom)

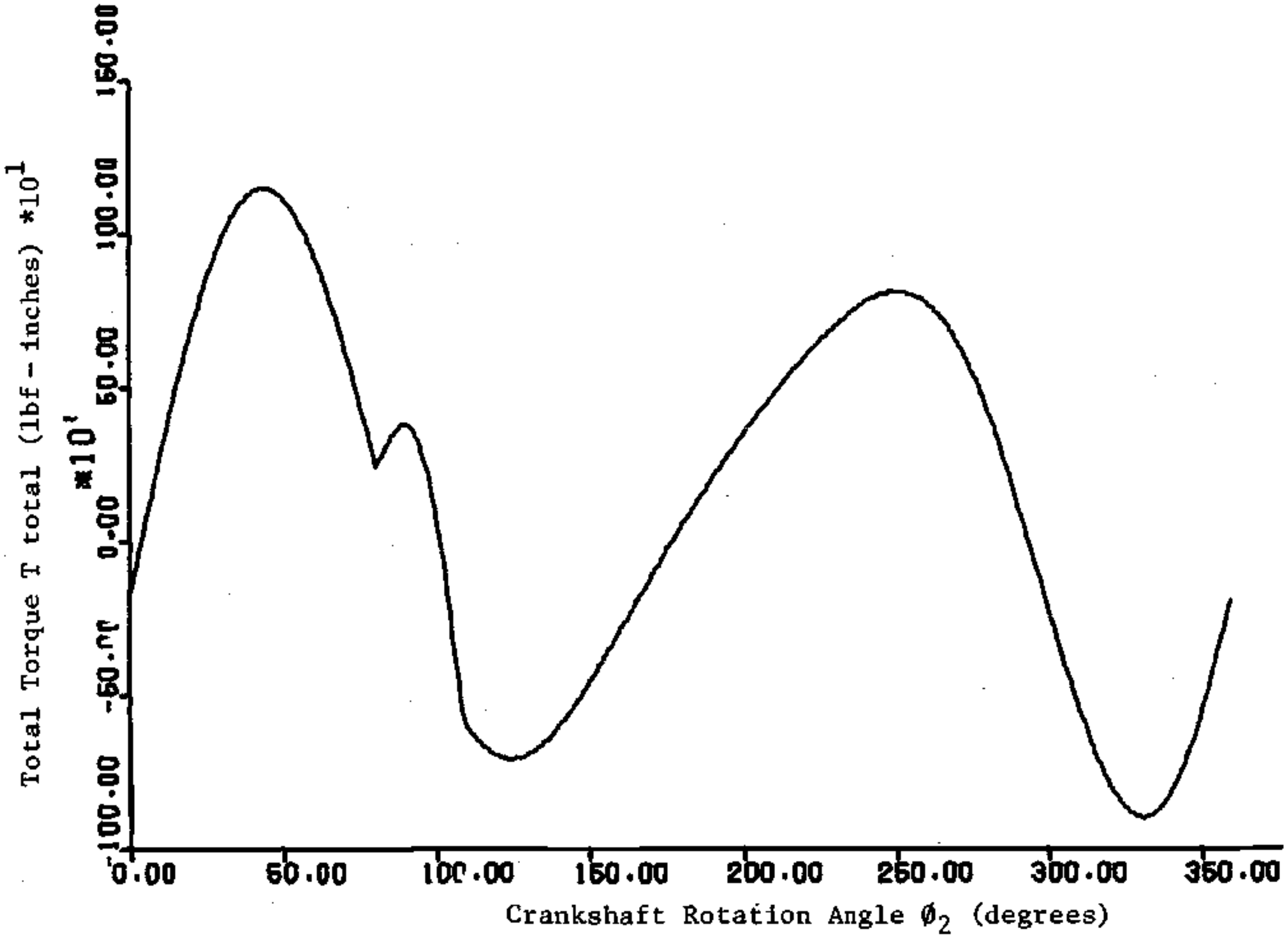


Figure 19. Total Torque Required T Total (lbf - inches) Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second (200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For Draper X - 2 Loom)

CHAPTER III

ENERGY CONSUMPTION MEASUREMENTS AND PROPOSED NEW MECHANISMS FOR THE FLY SHUTTLE LOOM

Energy Consumption Measurements

In order to verify the results obtained by analysis, a Draper X-2 Loom was run without a warp in many different ways. First of all, the loom was partially dismantled and the main bearings were cleaned out and relubricated. A watthourmeter, whose specifications are given in Table (6), was connected between the loom motor and the power supply. The average power consumed by the loom motor under any condition has found by using the following equation:

$$\text{Power (watts)} = \frac{K_h \ 3600}{t \text{ (seconds)}}$$

where $K_h = 24.0$ is the watthour meter constant and t denotes time in seconds required for the watthour meter disk to turn one revolution.

The energy consumed by the various mechanisms was measured by successive disconnection and addition of one or more mechanisms as shown in Table (7). Finally the actual power required to weave a medium weight fabric on an X - 2 loom was found by installing the power meter on another loom with a warp and running it. All the results are shown in Table (8).

Proposed New Mechanisms

The new proposed mechanisms reported in this thesis are only for

Table 6. Watthour Meter Specifications

Make	:	General Electric
Type	:	Two Element (V-2-S)
Amperes	:	50
K_h	:	24.0
Volts	:	240
Cycles	:	60 (3-wire)
Model	:	AL 251

Table 7. Status of Shafts, Mechanisms or Components of Draper X - 2 Loom for Each Experiment

Loom Shaft, Mechanism or Component	Status of Shaft, Mecahnism or Component							
	Experiment No.							
	1	2	3	4	5	6	7	8
Clutch	N	C	C	C	C	C	C	C
Crankshaft	N	C	C	C	C	C	C	C
Camshaft	N	C	C	C	C	C	C	C
Slay Mechanism	N	N	C	C	C	C	C	C
Picking Mechanism	N	N	N	C	C	N	C	C
Shedding Camshaft	N	C	C	C	C	C	C	C
Shedding Mechanism	N	N	N	N	C	C	C	C
Shuttle	N	N	N	C	N	N	C	C
Let off, take up warp and filling stop motions	N	C	C	C	C	C	C	C
Warp (Actual Weaving)	N	N	N	N	N	N	N	C

N = not ratating, disconnected or absent   C = rotating, connected or present

Table 8. Experimental Results

Erpt No.	Type of Operation	Wattage	% of Total
1	Clutch out	145.7	18.5
2	Two Main Shafts	288.0	36
3	Slay	454.0	57
4	Picking and Slay (with shuttle)	690.0	87
5	Picking and Slay (without shuttle)	635.0	80
6	Slay and Shedding	484.8	61
7	Shedding, Picking and Slay	720.0	91
8	Weaving	785.0	100

Total = 785.0 watts

x While conducting these experiments, the other loom motions like take up, let off and feeler were allowed to run at all times.

the beating up and picking motions. The shedding mechanism does not consume much power and also the torque required by it is relatively uniform.

The slay and picking mechanisms used in shuttle looms besides consuming the bulk of the power are also responsible for the large shaking forces, therefore generating vibrations and noise. Also, the fact that the slay and part of the picking mechanism are connected makes it sifficult to improve the operation of either mechanism beyond a certain point. Clearly, therefore, the logical approach would be to separate the two mechanisms. This idea has been used by designers for developing shuttleless looms, but, surprisingly, never in redesigning shuttle looms. One of the ways in which this can be accomplished is to run the slay mechanism with cams. Such methods have quite a few advantages. Firstly, the use of cams can made the slay operation intermittant and this can help improve the picking mechanism by allowing more time for shuttle flight. Secondly, the stroke of the slay mechanism can be reduced to a certain extent. Thirdly, the weight of the slay can be reduced by at least 100 lbs, and possibly more because of less stringent requirements. Such a beat up mechanism was theoretically investigated for a modified X - 2 slay. The modifications are listed in Table (9). The cam mechanism performs in a manner similar to that currently used in Draper double rapier looms (model DSL). The analysis of such a mechanism is similar to that of the shedding mechanism in Chapter II. The important dimensions are listed in Table (10). The torque versus input shaft rotation is shown in Figure (20). The average and R.M.S. power values

Table 9. Modifications of the Slay of the X - 2 Draper Loom for the Proposed New Mechanism

For calculation of the slay parameters for the proposed new mechanism the following parts have been removed from the X - 2 slay.

Part	Weight	X (inches)
Shuttle Boxes (left and right)	30.9	30.25
Check straps and their brackets (left and right)	18.2	26.5
Lug strap assemblies (left and right)	6.1	13.0
Picker stick and parallel motion assemblies (left and right)	41.75	3.8
Total Weight	96.94 lbf	

x = distance from center of rocking shaft to center of mass of the part

Table 10. Kinematic and Dynamic Parameters of Proposed Slay Mechanism

Parameters	Numerical Value and Units
c	11.75 inches
1	11.0 inches
B	90.0 degrees
γ	6.5 degrees
δ_o	15.0 degrees
k	0.0 lbf/inch
μ_o	0.0018
μ	0.15

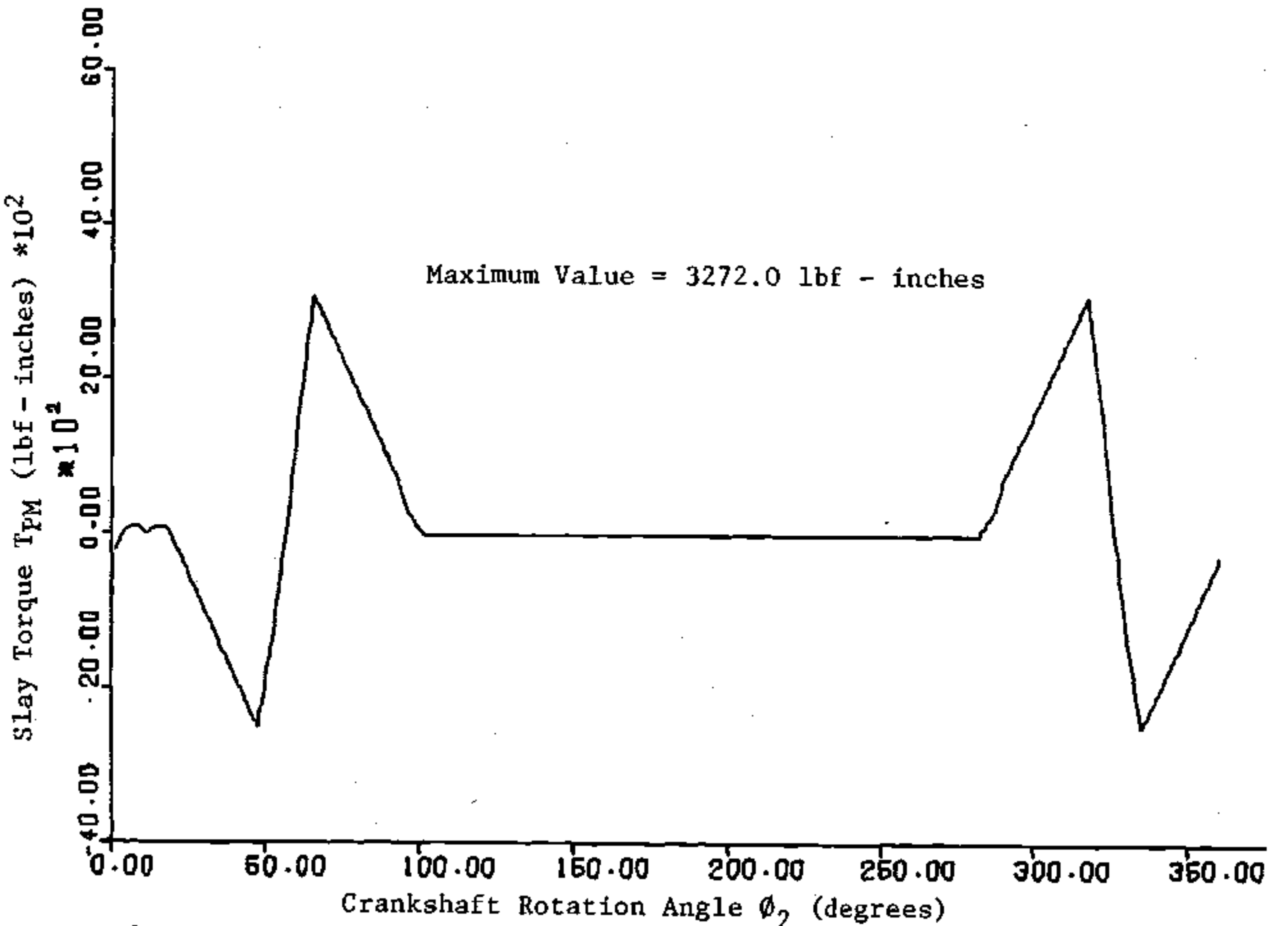


Figure 20. Slay Driving Torque T_{PM} (lbf - inches(Versus Crankshaft Rotation Angle ϕ_2 (degrees) [$\dot{\phi}_2$ = 20.95 radians/second 200 p.p.m., $\ddot{\phi}_2$ = 0.0 radians/second/second] (For proposed New Mechanism - 180° dwell)

are 300.56 and 1240.0 respectively.(watts)

For picking, an energy storage type of mechanism would be preferable from the point of view of energy use efficiency. The ratio of R.M.S. to average picking power is quite large and can be greatly reduced if the loom motor does not have to supply all the picking energy during a short time period. Such an energy storage and release picking mechanism has been proposed and developed by Tayebi[14]. A line diagram of this new picking mechanism is shown in Figure (21). In this particular example, the cam stretches the energy storage spring to its desired length and the picking release lever by virtue of its action transfers all the stored energy to the shuttle. Also, the inertia forces are minimized as only the picker, picker stick and parallel shoe have to be accelerated along with the shuttle. A restoration spring returns the stick to the proper position after picking. Such a system is therefore ideal from the point of view of energy use efficiency.

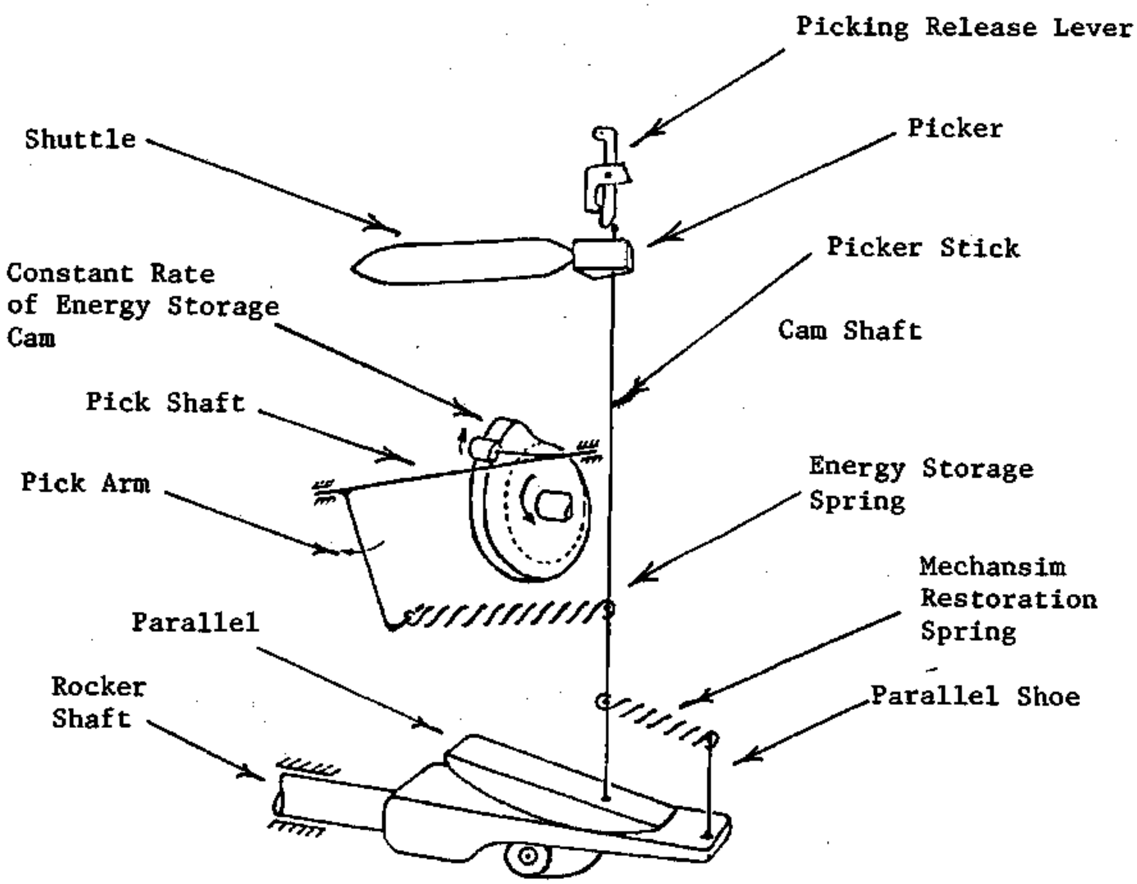


Figure 21. Constant Rate of Energy Storage Picking System

CHAPTER IV

DISCUSSION AND CONCLUSIONS

The slay mechanism is one of the major sources of vibrations and shaking forces, very definitely, it sould be improved and this can be done as the results in Chapter III indicate. A lighter intermittantly-operating slay would direct and indirect advantages. Some of the direct advantages are 2) smaller driving torque, b) lesser frictional losses in bearings, and c) longer picking time. All these advantages however are not evident from the results in Chapter III. This is partly due to the fact that the slay used is heavier than necessary from the point of view of rigidity and other requirements.

Also with improvement in the bearings used the frictional losses can be reduced and this would lead to better efficiency. The picking mechanism can be improved in many ways. The two methods proposed in Chapter III would lead to significant reduction in energy consumed as they would reduce the peak in torque demand, as shown in Figure (19), that is associated with any loom currently in use. By reducing this peak the change in torque demand as well as the rate of change in torque demand are reduced. These reductions are beneficial from the point of view of motor efficiency and power factor. Currently there is a 20% change (+10 to -10) in the average running speed of many looms as evidenced by Lord and Mohammed[13].

The shedding mechanism can be improved so that its dynamic behavior is better than what it is at present. Currently the shedding mechanisms in use have many flexible members and thus do not operate as well as they should. However, from the point of view of power efficiency, these mechanisms are quite adequate.

CHAPTER V

RECOMMENDATIONS

In the future, research efforts should be directed to improve both the slay mechanism and the picking mechanism, from the point of view of improvement in energy use efficiency and reduction in torque requirement. In particular, optimization studies should be carried out on different types of slay mechanisms by varying all the kinematic and dynamic parameters. Also, the different kinds of cam profiles suggested by Catlow and Vincent[5] should be studied in detail to improve the picking mechanism.

The principle of energy storage should be used both for the picking and slay mechanisms to improve their efficiency.

APPENDIX

APPENDIX A

REFERENCE SYSTEM FOR PLOTS

For all the quantities that have been plotted as a function of ϕ_2, the starting point has been taken to be that position of the crank as shown in figure (22). The 0° position of ϕ_2 corresponds to crank arm being parallel to the floor pointing in the forward direction. This position is very close to the beat up point but does not coincide with it. ϕ_2 is measured positive if the crank arm turns counterclockwise as shown in the figure.

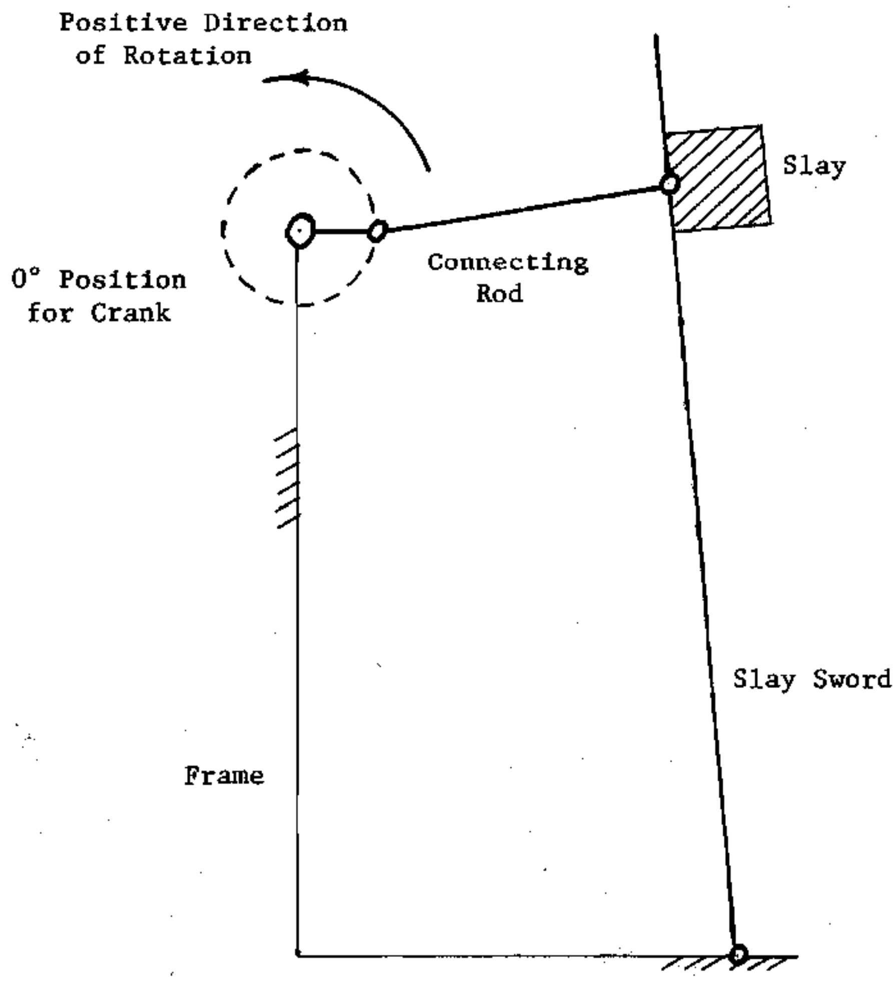
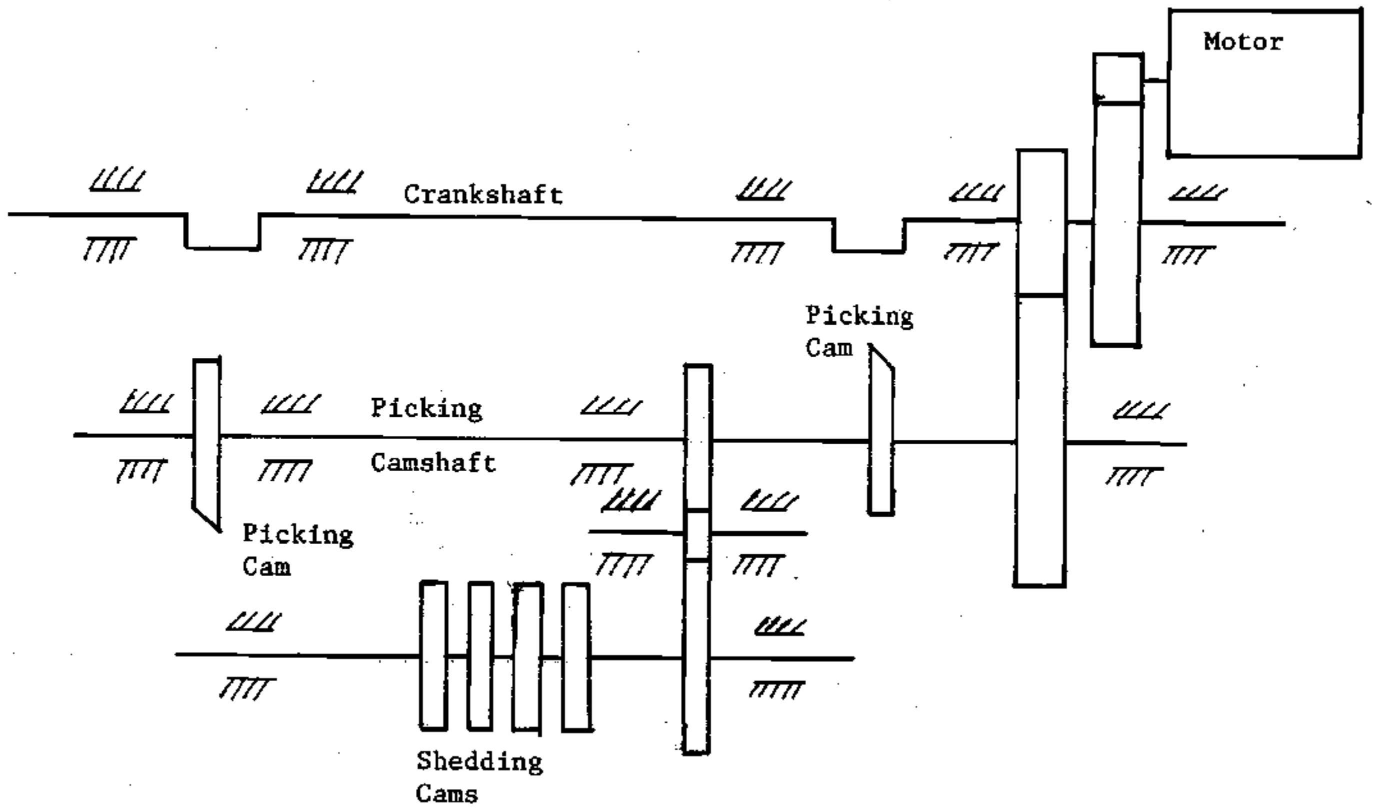
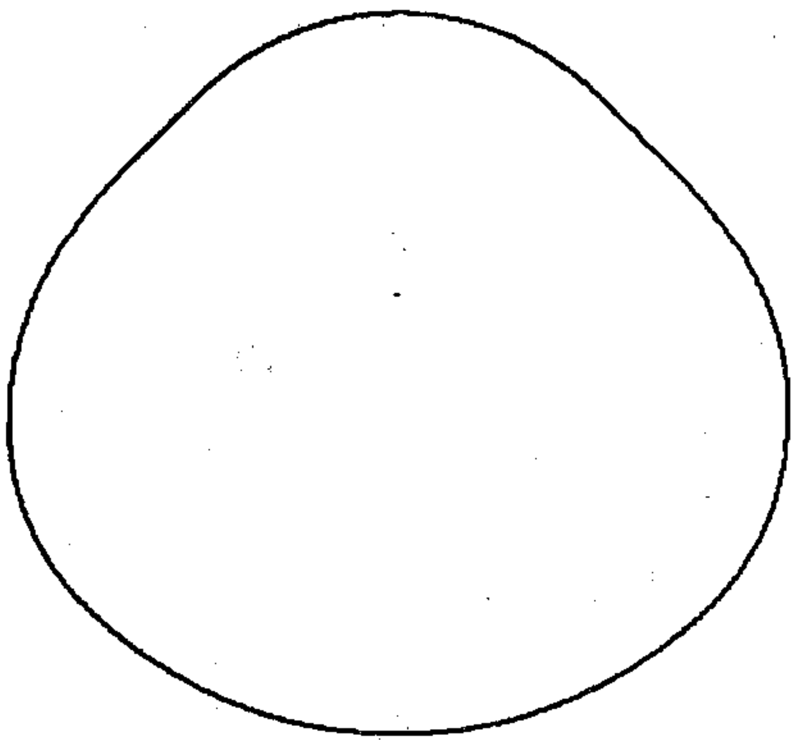


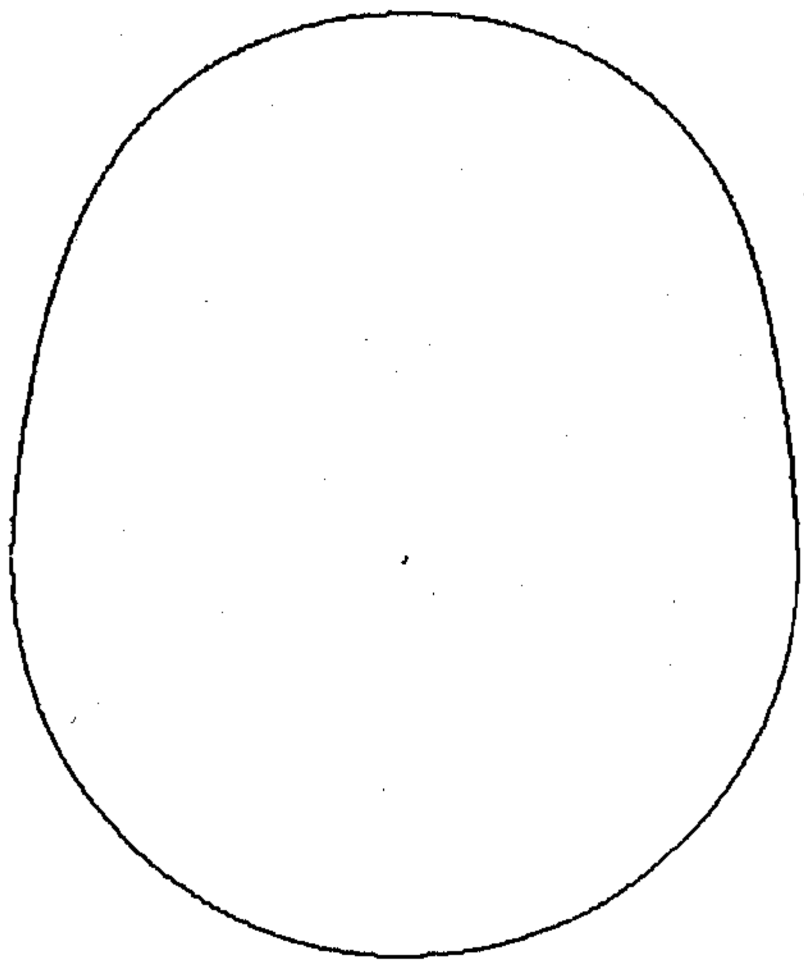
Figure 22. Reference System for Plots



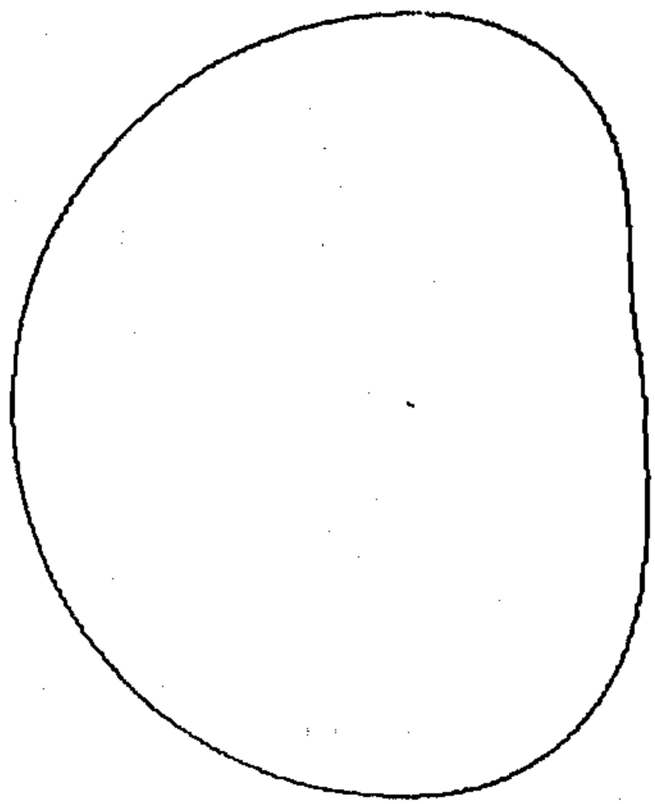
Appendix 2. Loom Gearing Diagram.



Appendix 3. Cam Profile for Shedding Cam
(Plain Weave - Draper X - 2 Loom)



Appendix 4. Cam Profile for Driving Cam (Intermittant Slay Operation - 180° dwell) (Proposed New Mechanism)



Appendix 5. Cam Profile for Receiving Cam (Intermittant Slay Operation - 180° dwell) (Proposed New Mechanism)

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